BUFFALO FAN SYSTEM

OF

Heating, Ventilating and Humidifying

CATALOG No. 700

Buffalo Forge Company

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FOREWORD

THE Buffalo Forge Company has always taken the stand that engineering data and developments should not be hoarded as hidden treasures but should be made available for the use and edification of the engineering profession in general.

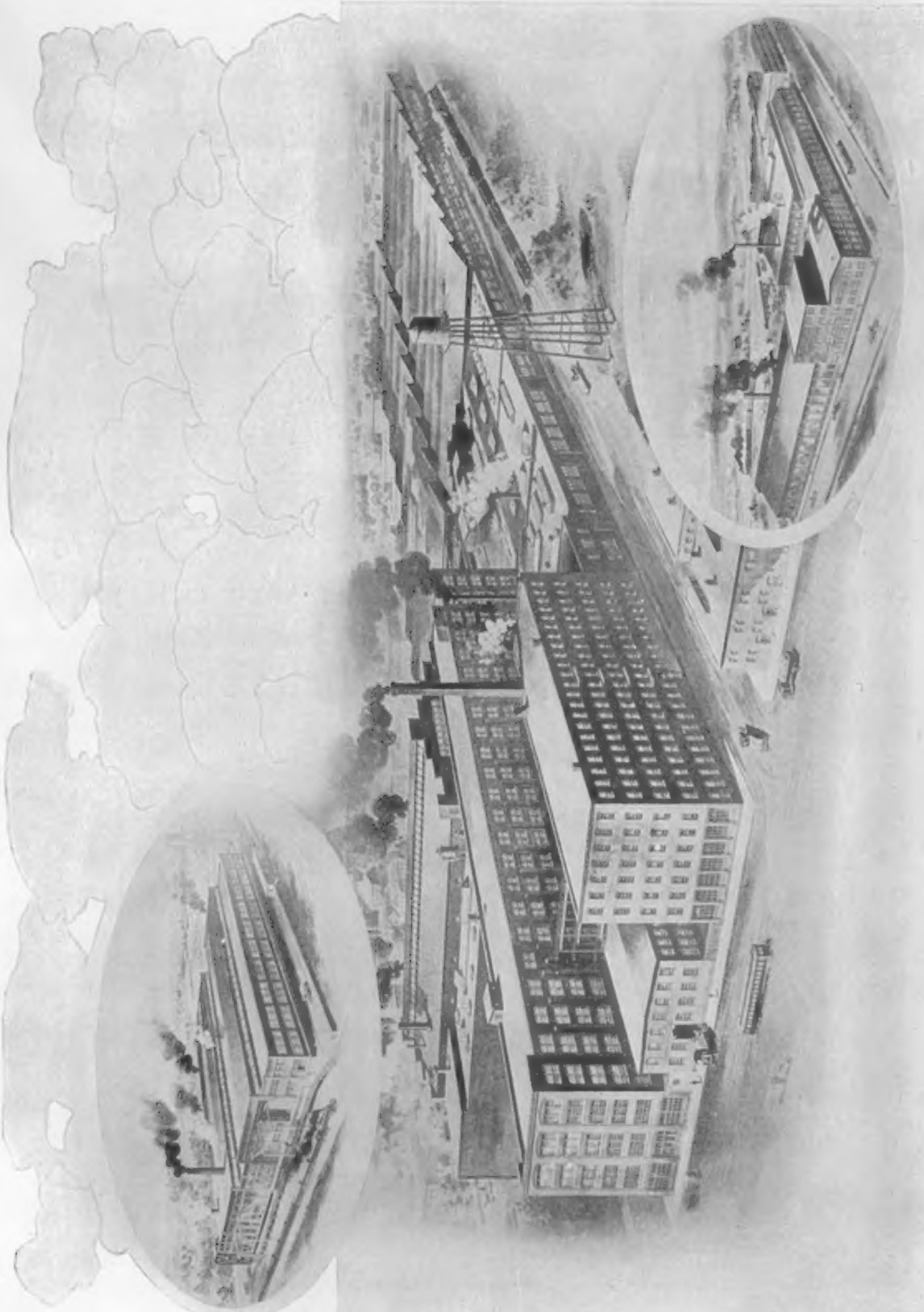
In this volume we have laid stress on the principles underlying all the various steps in the determination of suitable apparatus to meet all conditions of heating, ventilating and humidifying. These principles have been proven by actual practice and are the ones used by our own engineers in the solution of problems of a similar nature.

To the host of friends who gave our previous Catalogs Nos. 197 and 198 on Heating and Ventilating such a hearty reception we respectfully dedicate this volume.

Renew our acquaintance by letting our engineers help you with any problems you may have in Heating, Ventilating and Humidifying.

BUFFALO FORGE COMPANY

"Buffalo"



Canadian Blower & Forge Co.
Kitchener, Ontario

Buffalo Forge Company
Buffalo, New York

Buffalo Steam Pump Works
North Tonawanda, New York

THE BUFFALO FAN SYSTEM
OF
Heating, Ventilating and Humidifying

PART ONE

Public Buildings

IT has been within the last decade that the heating and ventilating art has come into its own. Many articles had been written on the subject and its importance insisted upon in theory, but unfortunately theory and practice had taken diverging paths. Through the earnest endeavors of the leading engineers, architects and physicians practice has now been made to accord with theory.

The following pages will serve not only to emphasize the importance of proper heating and ventilating but will describe such methods and apparatus as our engineers have used with great success in its attainment.

Years ago when our methods of living and working followed the natural lines and modes, usually those of least resistance, no need for ventilation other than by natural means was required. As our methods have become more artificial it has been found necessary to introduce artificial means to provide not only ventilation, but heating as well.

The progress of heating can be followed step by step from the rude fire of twigs down through the open fire place, the wood stove, and finally to the present day heating with steam, hot water and hot air. The development of methods of ventilation has been somewhat slower. The day when the opening of a window was ample ventilation has long since passed, and today we have grown accustomed to artificial means, such as fans, to supply positive ventilation.

Natural vs. Mechanical Ventilation

However there are very often times when some city official will fly up in arms and declare that the old style ventilation, that of the open window, is by far the best. It might be well at this point to give briefly the results obtained in a recent test. Taking a modern school, one half was ventilated by purely natural means, whereas the other half depended upon mechanical ventilation. Classes were conducted in the rooms under these conditions and observations taken at frequent regular intervals.

It was intended that these tests should cover the greater portion of one school year in order that all weather conditions might be experienced. The attitude of the teachers and pupils toward these tests was most favorable at the start and in

many instances certain teachers and pupils made their own choice as to whether they should be in naturally or mechanically ventilated rooms. Before the end of two months it was found necessary to discontinue the tests, this being due to the fact that teachers and pupils could not work to advantage in the naturally ventilated rooms and such stern objection developed that the tests could not be continued. The chief objections to natural ventilation were summed up by the impartial observers as follows:

1. "It was found impossible to keep the temperature and air motion conditions in the naturally ventilated rooms within the bounds of comfort.
2. The absence, because of illness of both pupils and teachers, in naturally ventilated rooms increased to an alarming extent.
3. The air in the naturally ventilated rooms was for the most part stagnant and heavy which caused depression and headaches."

Although these tests are not conclusive due to the short time over which they extended it is very plain to see that no logical arguments can be advanced for any comparison of natural with mechanical ventilation.

The more crowded a building is, the more complex becomes the problem of ventilation, for the exit and entrance of air must be complete and uniform throughout and at the same time all objectionable drafts must be avoided. With mechanical ventilation we are able to place the air just where it is needed and in just the right quantities and to remove all foul air as fast as it becomes objectionable.

Let us now consider what constitutes good ventilation and how it may best be attained.

Ventilation

In the human body, as well as in other animal organisms, life is sustained by a process of combustion in which the oxygen of the air is combined with the hydrogen and carbon of the food and carbon dioxide is formed as a result of this combustion. Therefore, a continuous supply of air with the proper amount of oxygen is just as essential to the sustaining of life as it is to the combustion of fuel under a boiler. We cannot, however, solve the proper amount of air to sustain life by any chemical formula, inasmuch as the "Livable" limit is reached long before the chemical limit.

The percentage of carbon dioxide in the air is a good indication of its state of purity but the exceedingly harmful effects of impure air are not entirely governed by an excess of carbon dioxide. Physicians have shown that the poisonous effect of respired air is due almost entirely to the organic matter exhaled from the lungs. The following table gives the comparison of pure air and respired air.

	Pure Air	Respired Air
Oxygen	20.35%	16.2%
Nitrogen	78.10	75.4
Carbon Dioxide	0.03 to 0.04	3.4
Water Vapor	1.5 Variable	5.

The respired air is immediately diffused in the air of the room and cannot be directly removed, therefore the air must be continually diluted till it ceases to be harmful. There is no definite standard of purity and any line drawn between good and poor ventilation is purely arbitrary. Pure air contains from three to four parts of carbon dioxide in 10,000. With an increase to 11 parts in 10,000 the air becomes noticeably opressive whereas an increase of three or four parts to a total

of six or seven parts is scarcely noticeable. Modern practice has been to consider good ventilation to exist when the air supply is so maintained that the total quantity of carbon dioxide does not exceed more than six to eight parts in 10,000.

It is estimated that the average adult at rest breaths 500 cu. in. of air per minute and exhales 17 cu. in. of carbon dioxide. From these figures we can determine the air supply necessary to maintain any standard of purity according to the following table.

Cu. ft. of air to be supplied per person for various standards of purity of air

Parts Carbon Dioxide in 10,000	Cu. Ft. air per min. per Adult	Per cent. of Respired Air
5	100	0.29
6	50	0.58
7	33.3	0.87
8	25	1.15
9	20	1.45
10	16.7	1.74
11	14.3	2.03
12	12.5	2.32

There are certain applications where more than the normal amount of air is necessary due to unusual conditions. The following table gives the air supply per person under various conditions.

Specifications of usual air supplied per person

	Cu. ft. per min.
Hospitals (Ordinary)	35 to 40
Hospitals (Epidemic)	80
Workshops	25
Prisons	30
Theaters	20 to 30
Meeting Halls	20
Schools (per child)	30
Schools (per adult)	40

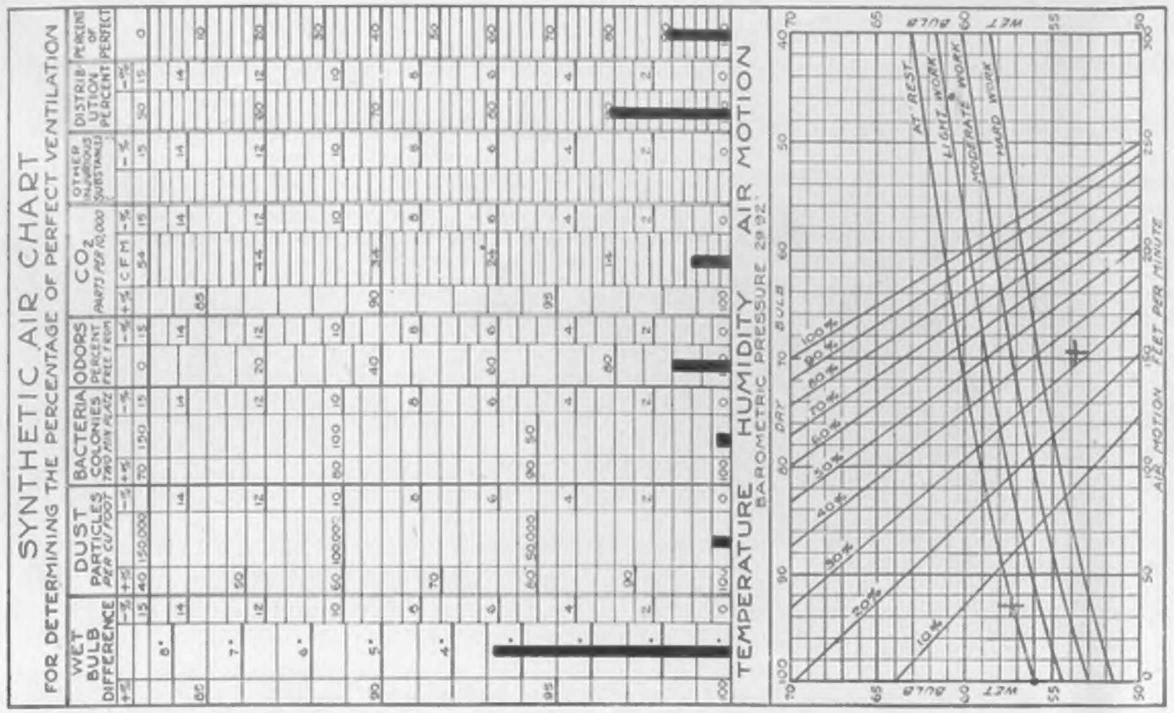
Dr. E. Vernon Hill has devised a very good method for determining the effectiveness or efficiency of ventilation.

This is done by using the Synthetic Air Chart shown on page 8 and we will quote Dr. Hill's explanation of its use.

The Synthetic Air Chart

"This chart is designed as a convenient method of recording data and arriving at a final percentage of perfect ventilation. It serves as a standard, or measuring "stick" as it were, for determining the efficiency of a ventilating equipment, and eliminates personal opinion and guess work. The chart includes all of the known factors that influence the ventilation of a room. They are as follows: Temperature and humidity, which are recorded as the wet bulb difference; dust, bacteria, odors; air supply and distribution as measured by the CO_2 content. These factors, furthermore, are each given their appropriate weight or value as a part of the whole. If all the factors are ideal the percentage as shown by the chart will be 100. If all or any one of the factors represent conditions that are not ideal the final percentage will be reduced in a corresponding amount.

After the results of a test are plotted on the chart we can see at a glance the final percentage of perfect, and if the results are not what they should be



TEST DATA

STREET No. 583 S STATE ST. DATE MARCH 1, 1918

BLDG JONES SCHOOL ROOM 10 FLOOR 2 TIME 9 A.M. TO 10 A.M.

STATION	TEMPERATURE DRY BULB	TEMPERATURE WET BULB	R.H.%	AIR MOTION	DUST	BACTERIA	ODORS	CO_2	SUPPLY REGISTERS AREA ⌀	SUPPLY REGISTERS VELOCITY	SUPPLY REGISTERS C.F.M.	EXHAUST REGISTERS AREA ⌀	EXHAUST REGISTERS VELOCITY	EXHAUST REGISTERS C.F.M.
1	70	54.0	34	35	5730	5	90	8.1	1.4	600	840	1.25	170	212
2	69	53.5	33	35		2	90	7.2	1.4	520	728	1.25	140	175
3	70	54.0	34	32	3680	5	90	6.1				1.25	130	163
4												1.25	140	175
5														
6														
7														
8														
9														
10	69.6	53.8	33.5	34	4705	4	90	7.1			1568			725

PRIMARY SENSE IMPRESSION FAIR NOTES ROOM 25'X30' 12' CEILING

No. OCCUPANTS	34	WINDOWS No.		RADIATORS		WEATHER FAIR
PHYSICAL STATE	AT REST	TYPE	D.H.S.	TYPE	SQ. FT.	TEMP DRY 26° WET 21° R.H. 40
AIR SPACE PER OCCUPANT	26.5	AREA	108⌀			WIND DIR 3 VEL. 6 MI PER HR
TOTAL AIR SUPPLY BY CO_2 PER MIN.	110.0	LEAKAGE	SLIGHT	WALL RAD	105	CHICAGO COMMISS'N ON VENTILATION
AIR SUPPLY PER OCCUPANT BY CO_2 " "	32.3	RATIO TO CU. CONTENT	1:83			TEST BY AEBERLY
" " " " " FAN " "	46.0	FLOOR AREA PER PERSON	22			PLOTTED BY T. WILSON
AIR DISTRIBUTION %	90.1					APPROVED BY E.V. HILL
						DRAWING No. 1256

the factors that reduce the final percentage are at once determined. The longer the test line in any column the less favorable are the conditions represented. When the test line disappears conditions are perfect.

Under each factor there are three columns, a plus percentage column; the factor column proper, and the minus percentage column. The factor column proper is divided into appropriate units of measurement, as degrees for temperature, particles per cubic foot for dust, colonies for bacteria counts, etc.

The plus percentage is the percentage of perfect for the specific factor in the column; for example, in the chart shown in the illustration the dust count is 5,000 particles per cubic foot. This gives a plus percentage of 98, meaning that so far as dust is concerned the air is 98% free.

Reading the minus dust column we find the percentage as a part of the whole chart is only one-half of one per cent. This one-half of one per cent., together with the other minus percentages from the various columns, is deducted from 100 in arriving at the final percentage for the entire test.

The curves at the bottom of the chart headed 'Temperature, Humidity and Air Motion' are for determining the wet bulb difference. To do this proceed as follows:

Mark a point on the curve indicating the wet bulb temperature determined by test. This point should be located at the intersection of the wet and dry bulb lines. This is done as a matter of convenience, as the point will then give the wet bulb, the dry bulb and relative humidity.

Next mark by a point on the line denoting the physical state of the occupants, the air motion from the test. This point will be at the intersection of the appropriate physical state curve designated by 'At Rest,' 'Light Work,' 'Moderate Work' and 'Hard Work' with the vertical line of air motion. The vertical distance between the two points is the wet bulb difference, that is, it is the variation in degrees between what the wet bulb should be and what it actually was by test. This wet bulb difference is plotted in the first column of the chart.

The distribution factor is the percentage of distribution in the room. It is determined by an analysis of the air samples at various points for CO_2 and the average of all samples taken is the average distribution for the room.

The percentage of distribution is the percentage of variation of the different samples from the average.

The reverse side of the chart, illustration No. 2, is arranged for recording test data."

HEATING

Closely associated with the problem of proper ventilation is that of satisfactory heating, in fact it is very hard to draw the line of separation between the two problems.

Room Temperature

The physical principles involved in heating buildings are more complex than usually supposed, and exhibit an admirable nicety in the balancing of forces.

The first factor to be considered is the heat generated by the human body and the methods for its disposal. These are important conditions which determine the most desirable room temperatures and in densely peopled buildings, largely determine the result of vital processes dependent in part upon the activity of the individual. This amount of heat extends over considerable limits as shown by the following table.

Child six years old	240	B. T. U. per hour.
Adult at rest	380	" " "
Adult at work	500	" " "
Man 30 years old in an atmosphere with a temperature of 68° F.	400	" " "
The same in a atmosphere of 31° F.	600	" " "
Woman 32 years old	480	" " "
Adult in old age	360	" " "

We have found that a good average value for the amount of heat in B. t. u.'s given off per person per hour in an atmosphere of 70°F. is 400 for adults and 200 for children, these figures being generally used when the heating of densely peopled buildings such as schools and auditoriums, is considered. The average normal temperature of an adult in health is 98° F. and since heat is continually being generated, it must be disposed of as fast as generated. This disposal may be accomplished in three ways:

First: By direct transmission or radiation to the surrounding air.
Second: By the absorption of heat in the evaporation of perspiration.
Third: By the evaporation of moisture through the lungs.

Radiation depends upon the difference in temperature between the body and the surrounding air, but it is also affected by the amount of clothing and the humidity of the air. It is evident that the temperature of the room should not be so low that the body will radiate more heat than produced under normal conditions. In fact, from a hygienic standpoint, less heat should be absorbed than is generated, thus allowing part of the heat to be absorbed by perspiration. The following room temperatures have been found to give the best results.

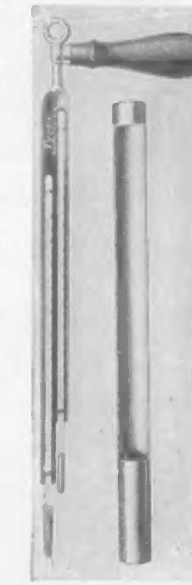
Public buildings	68 to 72° F.
Machine shops	60 to 65° F.
Foundries, boiler shops and all places where physical labor is done	50 to 65° F.

Heat Losses

In order to maintain a fixed temperature within a building, it is necessary to supply heat in sufficient quantities to compensate for the heat loss through the walls and roof of the building, and also to heat up the outside air brought in for ventilation. The subject of heat transmission in buildings has been thoroughly investigated so that the laws governing it and the factors for heat transmission of various building materials and building constructions are now quite accurately known. It has been found that the loss of heat by transmission is proportional to the difference of the temperature on the two sides of the material. The table on page 109 shows the accepted factors of heat transmission.

Humidity

The amount of moisture that a given quantity of air can hold increases very rapidly with the temperature. The amount per cubic foot of air is the measure of its humidity and this humidity has a great bearing upon livable conditions in schools and public buildings. It may be to advantage to explain at this point how the relative humidity of air may be determined. If a thermometer bulb is covered with a damp cloth a drop in apparent temperature of the surrounding air will ordinarily result. This temperature is called the sensible or wet bulb temperature. The less the humidity the greater the difference between the actual and the sensible temperatures, while at 100% saturation they are the same.



To determine the wet bulb and dry bulb temperature a sling psychrometer is used. This is clearly shown in the cut and consists of two thermometers, one having the bulb covered with a thin gauze. In taking the readings the gauze is moistened and the psychrometer is then rapidly whirled around until the mercury readings in the two thermometers stay constant. It is essential that the instrument be moved rapidly or held in a current of air for an air movement across the wet bulb is necessary to obtain a true reading.

From the hygienic standpoint it is evident that the means for regulating the humidity is just as important as the problem of proper ventilation and proper heating in every school and public building.

Psychrometric Charts

The relation between the temperature as measured by the wet and dry bulb of air and the moisture content is clearly shown in the psychrometric chart on page 13. It will be seen that a cu. ft. of air at 70° will hold eight grains of moisture while at 32° it will hold only two grains and at zero only five-tenths of a grain. The normal limits of humidity vary from 50% to 75% of saturation. It has been found that when the humidity goes above or below these points the condition becomes very uncomfortable and in fact injurious to health. Hence, it will be seen that air at 70° should contain from four to five and one-half grains of moisture per cu. ft. to be in the best condition for ventilating purposes. In the ordinary methods of heating with the air temperature 32° outside, the humidity of this air when heated to 70° without the addition of any moisture would be only 15.5% which is far less than the humidity of the driest climate known. It is this extreme dryness of the air in a heated room which produces the commonly noticed discomforts, such as extreme thirst, a parched feeling in the nose and throat, lassitude and headache. This extreme dryness has been a contributing source to many throat and pulmonary diseases.

The Psychrometric Charts on pages 16A and 16B are taken from the catalogue of the Carrier Air Conditioning Company of America, one of our associates in business. These two charts should be used when calculations are made in terms of pounds of air, while the chart on page 13 should be used when the pound cubic foot is a unit. For most purposes of calculations it will be found preferable to use the pound as a unit.

The various curves shown on these charts will be found especially valuable in making air calculations. The grains of moisture per pound of dry air are read by passing directly from the dew-point, or intersection of the wet- and dry-bulb temperatures, to the scale on the left edge of the chart. The B. t. u. required to raise one pound of dry air one degree when saturated with moisture, as also the vapor pressure, may be determined by passing vertically from the dew-point to the proper curve, and then to the corresponding scale on the left edge of the chart. The total heat, in B. t. u., above zero degrees contained in one pound of dry air saturated with moisture may be found by passing vertically from the wet-bulb temperature to the total heat curve and then to the left edge of the chart. The volume of air in cubic feet per pound may be found by passing vertically from the dry-bulb

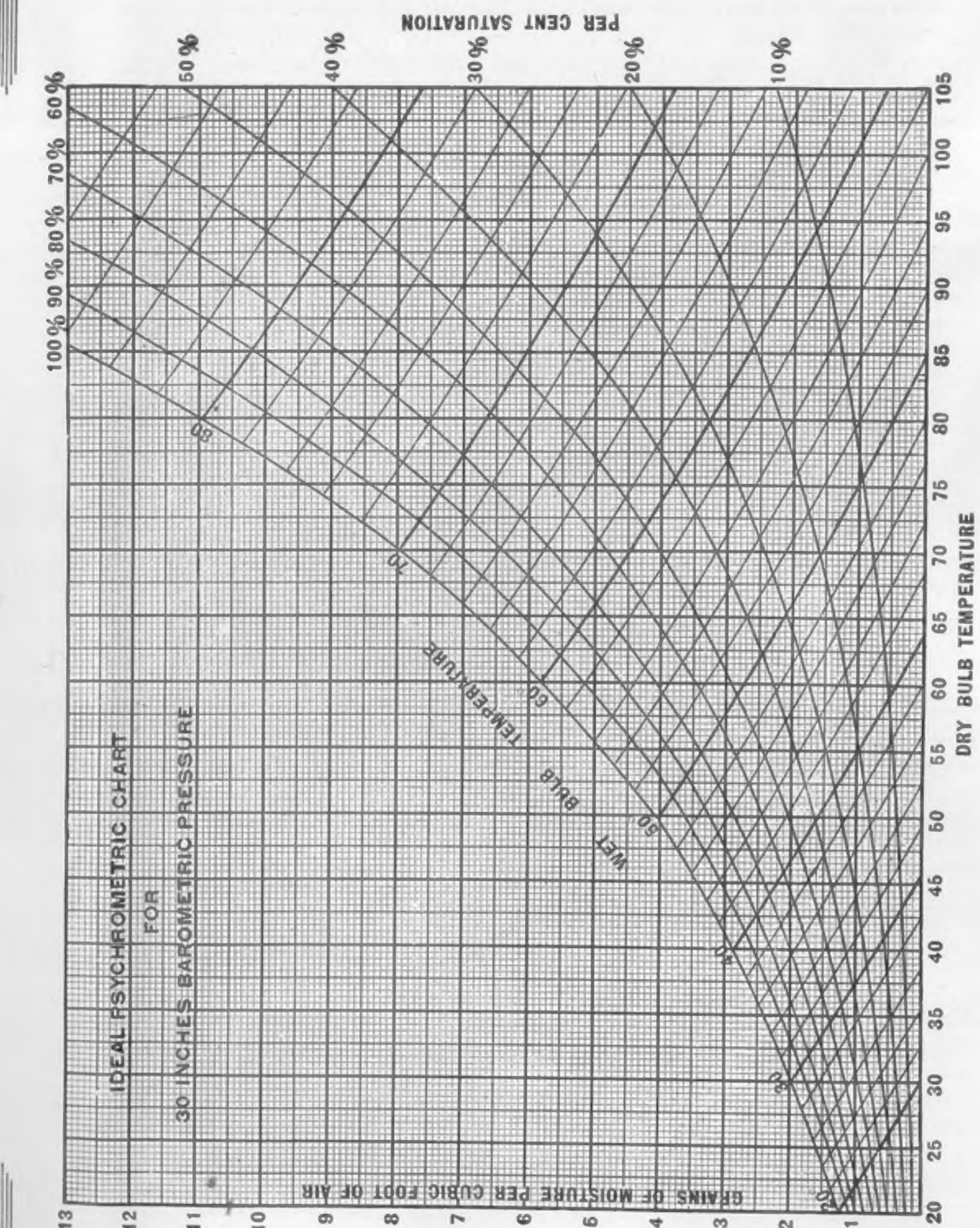
temperature to either of the two volume curves and then to the left edge of the chart. One curve gives the volume of dry and the other of saturated air.

Example. As an example of the use of this chart we will assume air at 75° dry-bulb temperature and 60 per cent. relative humidity. From the chart we find that the wet-bulb temperature will be 65.25°, the dew-point 60°, the grains of moisture per pound of dry air 77; the heat required to raise one pound of dry air saturated at 60° through one degree is 0.24664 B. t. u.; and the vapor pressure of air saturated at 60° is 0.523 inches of mercury. Passing vertically from the wet bulb temperature of 65.25° to the total heat curve and thence to the scale on the left, we find the total heat above zero in one pound of dry air when saturated at 65.25° to be 29.75 B. t. u. This, then, is also the measure of the heat in a pound of air at 75° and 60 per cent. relative humidity, since the wet-bulb temperature is the same.

The cubic feet per pound of air may be found by passing vertically from the dry-bulb temperature to either of the two volume curves, depending on whether the volume of dry or of saturated air is desired. To determine the volume of one pound of partly saturated air as here assumed, we will have from the chart.

Cu. ft. per lb. at 75° sat. =	13.88
Cu. ft. per lb. at 75° dry =	13.48
	.40 = Moisture
	.60 = Rel. Humidity
	.24
	13.48
Cu. ft. per lb. at 75° and 60% =	13.72

As an example of the use of the chart on page 13 we will assume a case where the dry-bulb temperature is 80° and the wet-bulb thermometer reads 70°, or a 10° depression. From the intersection of the corresponding lines through these two temperatures we find the relative humidity to be 62 per cent. Passing horizontally to the left from this point of intersection to the wet-bulb temperature line (called the saturation curve) we find the dew-point temperature to be 64.5°. If the temperature of the air should be reduced both the dry- and wet-bulb readings will be lowered until they both read 64.5°, when the air will be saturated. The grains of moisture contained in each cubic foot of this air will be found by continuing to the left on the horizontal line through the 64.5° dew-point to the left edge of the chart, where we have a reading of 6.65 grains. If the temperature of the air be further reduced, part of the moisture content will be condensed, the dew-point or saturation temperature will be lowered, and the grains of moisture per cubic foot will be correspondingly less.



Methods of Heating, Ventilating and Humidifying

The old fashioned fire place was the first attempt at heating and ventilating. The draft produced by the large chimney gave ample ventilation, but the heat loss along with this ventilation was very large and hence, as a heating system, the open fire place was most uneconomical. The next step, the old stove, afforded practically no ventilation, although its economy from a heating standpoint was fairly high. The modification of the old stove, namely, the hot air furnace afforded a certain measure of ventilation but this measure was far too limited and unreliable to make its use permissable in large or crowded buildings. A serious objection to the hot air heater is the liability of coal gas leaking into the air. The hot air furnace is the chief offender in heating with extreme dry conditions of air as described in the paragraph on humidity on page 11.

The next step is marked by the introduction of direct radiation with steam or hot water furnaces. Owing to its cheapness this method has been extensively introduced but it provides for no ventilation other than by windows and doors, and the resulting close, stuffy, heated rooms in office and other public buildings have doubtless increased materially the world's death rate.

The use of indirect radiation permits a certain amount of ventilation and elaborate systems have been devised on this basis. Aspirating shafts for removing foul air in connection with indirect systems have given positive results. In the latter system radiators are placed in the ventilating shafts to produce a draft by increasing the temperature of the foul air. The cost of ventilation by this method is expensive and the use of aspiration flues as substitutes for fans is indefensible.

The Buffalo Fan System

It is today universally acknowledged that the fan system has solved the problem of the successful heating and ventilating of public buildings. In recognition of this the legislatures of practically all states have passed statutory laws, requiring the use of the fan system of ventilation in school buildings. The question no longer is "Shall the fan system be used?" but "How may it best be applied?"

For the past 35 years the Buffalo Forge Company has been engaged in designing heating and ventilating systems and in the construction of such equipment. This company has its systems in successful operation in thousands of buildings in this country, Europe and Japan, in fact in all parts of the civilized world. The improvements put forth by this company have brought the art of heating and ventilating to a degree of perfection not previously known. One of the most important of these improvements is the Carrier Air Washer and Humidifier, which removes all impurities from the air and imparts to it the proper humidity.

Public Buildings

There are two arrangements of the Buffalo Fan System as applied to public buildings.

The first, in which the fan system handles both the heating and ventilating requirements and the second, sometimes called the split system, in which the heating requirements are taken care of by direct radiation in the room and the fan system handles the air required for ventilation only.

The apparatus used in the two applications differs only in the amount of heater surface required. The equipment consists of a boiler for the generation of steam, a centrifugal fan, driven by an engine or motor, for the propulsion of air, an air washer for purifying and humidifying, a steam radiator for heating and a system of ducts for distributing the heated air and for removing the foul air.

The boiler may be of any customary type and may be operated at any pressure between one-half and 100 pounds per square inch, however, a pressure of 20 pounds is most desirable when a steam engine is used as the prime mover for the installation.

The fan is of the centrifugal type and is usually constructed as an exhauster, i. e., with only one inlet. The use of a steam engine as the prime mover allows for great economy since the exhaust steam can be utilized in the heater, this greatly reducing the cost of power used. The Buffalo heater described in detail on pages 52 and 53 consists of vertical coils of one inch full weight steel pipe screwed in cast iron manifold bases. Steam is supplied to the coils on one side of the manifold and exhausted from the other side, both the inlet and exhaust connections being on the same end of the base. Separate steam and exhaust connections are provided for each of the several sections into which the heater unit is divided, each connection being supplied with valves allowing as many or as few heater sections to be in operation as are needed.

The fan may be placed so that the fresh air is either forced through or drawn through the stacks of heater coils. In public buildings it is the general practice to separate the heater into two parts, one part known as the tempering coils, containing from six to ten rows of pipe, the amount being just enough to heat the incoming air to a temperature of from 60° to 70° before it reaches the washer or fan; the other part known as the heater proper is placed at the fan outlet. The size of the heater is governed by the amount of air to be handled and the temperature required on leaving the fan. Between the tempering coils and the fan is placed the Carrier Air Washer and Humidifier. The air, after being tempered, cleaned and humidified is discharged under pressure into two chambers known as the hot and tempered air plenums, respectively. In the hot air plenum chamber are placed the heater coils, while the supply to the tempered air plenum is carried by a by-pass either above or underneath the heater. In the split system no by-pass around the heater is necessary. After leaving the heater the air is distributed by means of ducts leading to the room to be heated and ventilated.

It is customary to place the outlet registers so that the heated air enters the room about eight feet above the floor, this height being sufficient to prevent drafts and still allow for the proper air velocities through the registers. The cold or foul air is removed by vent registers placed at the floor line usually on the same side of the room as the hot air flue. The heated air enters above at a higher temperature than that of the room, and a complete and practically uniform diffusion to all parts of the room occurs. The cooling effect of the outer walls and windows produces a downward circulation at these points with a consequent flow from the hot air registers toward the outer wall in the upper stratum, and a flow from the outer walls to the vent registers in the lower or breathing stratum. This flow occurs over such large areas that the velocity is most imperceptible.

Inasmuch as the heated air is positively supplied to the room, the foul air must be positively forced out through the only channels available, namely, the vent register, flues, and leakage cracks around the windows and doors in the outer walls.

Exit by the latter means is necessarily restricted in properly constructed buildings, but it serves the purpose of preventing the undesirable infiltration of cold air which would otherwise occur. The above method is often spoken of as the plenum system.

It is just as important to positively remove the foul air as it is to positively introduce the fresh air but the same progress has not been made in this end of ventilation. The usual method is to have vertical flues with roof ventilators and depending entirely upon the stack effect to remove the foul air.

Many of the best installations provide a supplementary fan system for exhausting the foul air.

Advantages of the Buffalo Fan System

The contrast between the methods and effects of the old system of direct radiation depending upon windows and doors for ventilation on one hand, and the Buffalo Fan System of Heating and Ventilating on the other is very striking. With direct radiation all air for ventilation must be admitted at the windows through the lower sash. This is made necessary because any opening of the upper sash will allow the escape of the stratum of heated air in the upper parts of the room. This method is both unsanitary and uneconomical. It is unsanitary, first, because it is impossible to admit sufficient fresh air by means of windows without objectionable drafts; second, the ventilation is not uniform, and depends entirely upon atmospheric conditions outside the room, being mostly affected by the direction and velocity of the prevailing winds; and third, an undesirable layer of cold air tends to settle along the floor, which does more harm than an entire lack of ventilation. It is uneconomical because the coldest air remains along the floor, and the heated air rises and flows out of the window openings. The heated and cold air do not get an opportunity to intermingle and most of the heat produced is not used to advantage. Further the heat is not equally distributed, the better ventilated parts of the room are too cold, and poorly ventilated parts are too hot; the room temperature cannot be kept uniform or regulated to any extent and the loss due to overheating is great.

The Buffalo Fan System, on the other hand, is sanitary and economical and overcomes all the objections voiced against natural ventilation, because it maintains a uniform temperature, prevents all drafts and secures a warm floor. It is economical, because the temperature is readily and absolutely controlled either automatically or by hand, and any overheating is prevented. This latter advantage is very much greater than is generally supposed.

Carrier Air Washers

One objection that is frequently raised to the use of the fan system is that dust is drawn in with the air and blown into the room. This objection can be very easily overcome by the Carrier Air Washer which positively removes all traces of dust, soot and smoke, and the foulest germ-laden air of the city is thus made as clean and pure as that of the country. The advantage of this process wherever cleanliness and sanitary conditions are desired is easily appreciated and renders this system particularly valuable in libraries, hospitals, schools, in fact in all buildings where clean air is a requisite.

These pails contain dirt, mud, soot, bacteria of various sorts, and disease-breeding filth of all kinds which was washed from the air used for ventilation of Public School No. 6, Brooklyn, New York, and shows the result of one week's run of the Buffalo Fan System.

This mud was shoveled from the bottom of the Carrier Air Washer settling tank after the water had been drained off. Of course all the finest dirt floating in the water had been carried off.

Had it been possible to strain the water as it was drained, no doubt five more pails would have been filled. These pails each contained approximately twenty-five pounds of dry dust so this washer was collecting approximately one hundred and twenty-five pounds of dirt-carrying disease every five days.

Another big advantage of the air washer is that the humidity of the air used for ventilation can be positively controlled. The advantage of this has been described under the subject of humidity on pages 10 and 11.

Humidity Control

The Carrier Air Washer and Humidifier described on page 50 effectively overcomes the dryness of ordinary heated air and places the humidity of the air under accurate and automatic control. In this system the humidity of the air entering the building is regulated to the finest nicety through the control of the temperature of the spray water.

This method of regulation is the only simple and direct form of humidity control. By means of the spray water temperature regulation every demand for variation is immediately taken care of and there is no delay between the demand and response as is evident in all other methods of regulation. The temperature of the spray water is raised by the introduction of steam through a device similar to an injector, or by a closed water heater.

It has been proven by numerous tests that the temperature of the spray water is a greater factor in the amount of moisture which the air will absorb than the temperature of the air itself.

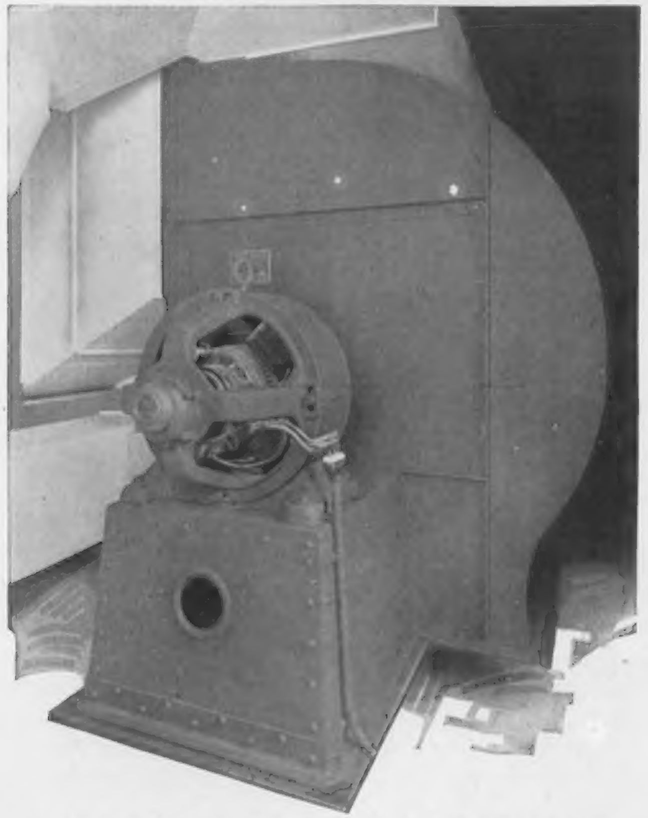
Regulation of the temperature of air entering the ventilating system as a means of controlling the amount of vapor absorbed is inadequate, and attempts to secure a constant relative humidity by regulating the temperature of the body of water in the settling tank of the air washer fail on account of the time element before this water is sprayed into the air. In these and other systems two thermostats jointly aim to give the control desired, bringing in a double error, and a considerable lag of regulating effect behind the outside atmospheric changes which cause it.

The Carrier Dew-point System of humidity control uses one thermostat of very simple and accurate design exposed to the temperature of the washed air, controlling directly the temperature of the water as it is sprayed, not of the whole

volume of the settling tank. There is no lag, cause brings instant effect, and literally any relative humidity may be maintained automatically.

Reduction of the humidity is not desirable except for special processes in the industries, but may be accomplished by the use of refrigeration for cooling the spray water. The average winter temperatures in our Northern States set a practical upper limit for humidity at 40% to 45% above which the coldest weather will cause condensation on windows. See discussion on page 30.

The Dew-point System indicates by its name that the air must be saturated, thus fixing absolutely the number of grains of moisture per cubic foot at a given temperature which leaves only the temperature of the saturated air to be controlled. No air washer that will not give saturation can be used with the Dew-point System, but Carrier Air Washers have spray systems which make saturation possible when using heated spray water.



Buffalo Ventilating Unit in Bowery Branch Y. M. C. A., New York City

Hospitals

The necessity of ample ventilation in hospitals is not receiving the proper attention by those most concerned. Although absolute cleanliness is paramount in the mind of the physician it is really surprising that this question is so frequently lost sight of when hospital ventilation is considered. This matter is being brought forward by the leading engineers and is gradually coming into its own.

The extreme importance of maintaining the proper humidity in the treatment of certain diseases is just being realized by physicians. In diseases of the heart and the respiratory organs, in fevers and especially in all nervous disorders, patients are extremely sensitive to changes in humidity and adversely affected by the dryness of the air ordinarily existing in heated buildings.

When used for cooling the hospital rooms in the heat of summer the Carrier Air Washer in connection with the Buffalo Fan System proves efficacious and convenient.

Libraries

The Buffalo Fan System in connection with the system of air purifying is applied to the heating and ventilation of library buildings with most satisfactory results. Not only does it afford positive ventilation, but it frees the air from all traces of smoke and dust so objectionable in libraries; besides, outward pressure of air in the building prevents the entrance of dust from without.

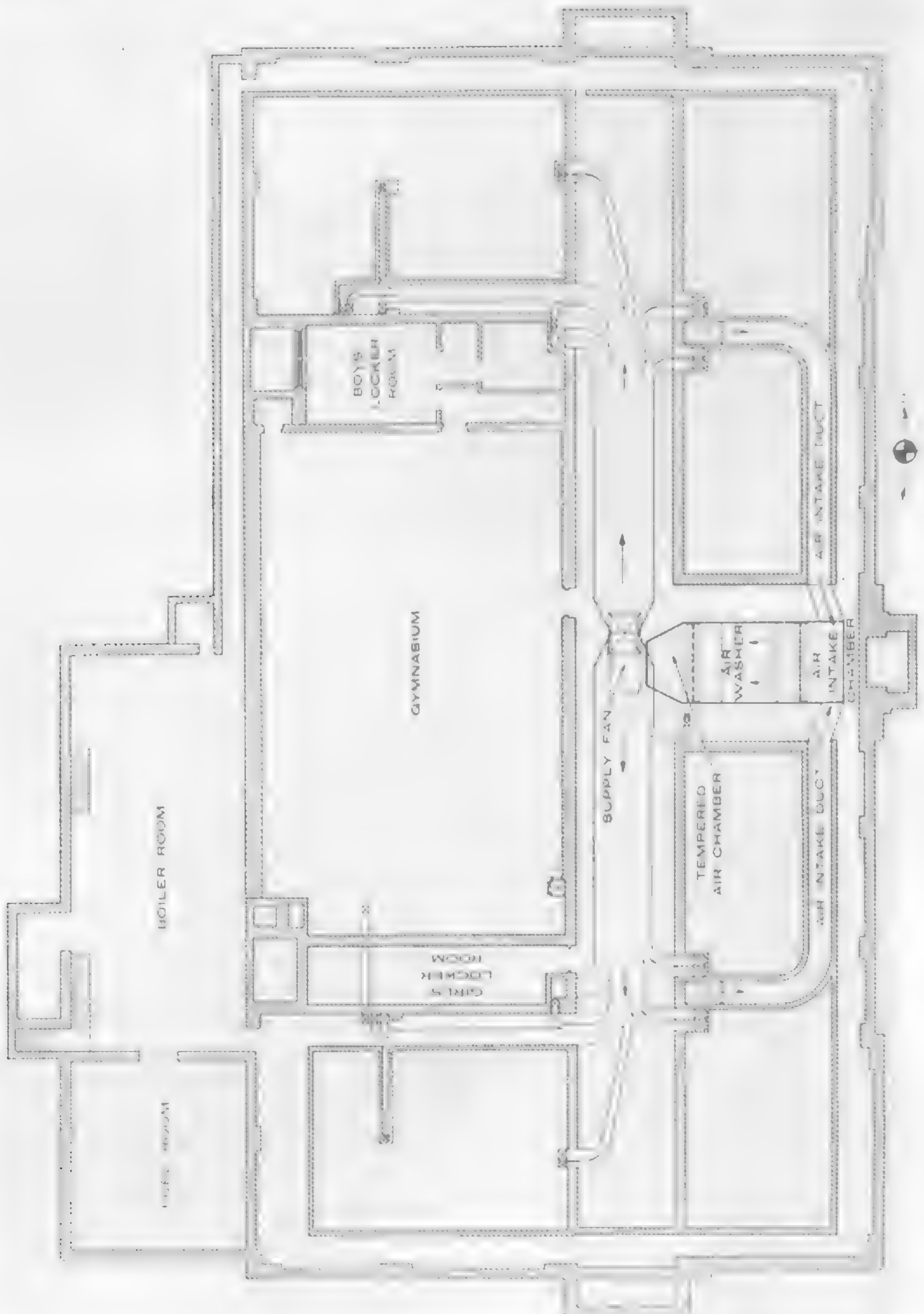
In one instance tests were made of the temperature of the water in circulation and of the air at various points with the results shown in the following table:

Carnegie Branch Library, St. Louis, Mo.

Room	Time P. M. 2:30	2:50	3:15
Auditorium, basement	75	75	74
Stall room, basement	79	80	77½
Stall room, basement	77	77	76
South reading room, main floor	78	78	78
North reading room, main floor	78	78	78¼
Stock room	79	80	79½
Average	77.7	78	77.2
External air			86
Air entering rooms			73
Circulating water			69

It is interesting to note the effect of this apparatus in cooling the building. Although the temperature of the external air was 86° F., it entered the rooms at 73, and kept their temperature down to between 77 and 78, a cooling of about 8½°.

During the first two series of readings the windows in the three basement rooms were open. Before taking the 3:15 P. M. reading they were closed. The result shows that the temperature of these rooms was noticeably lowered by excluding the external air and supplying only the washed air from the fan.



School Layout

Application to Schools

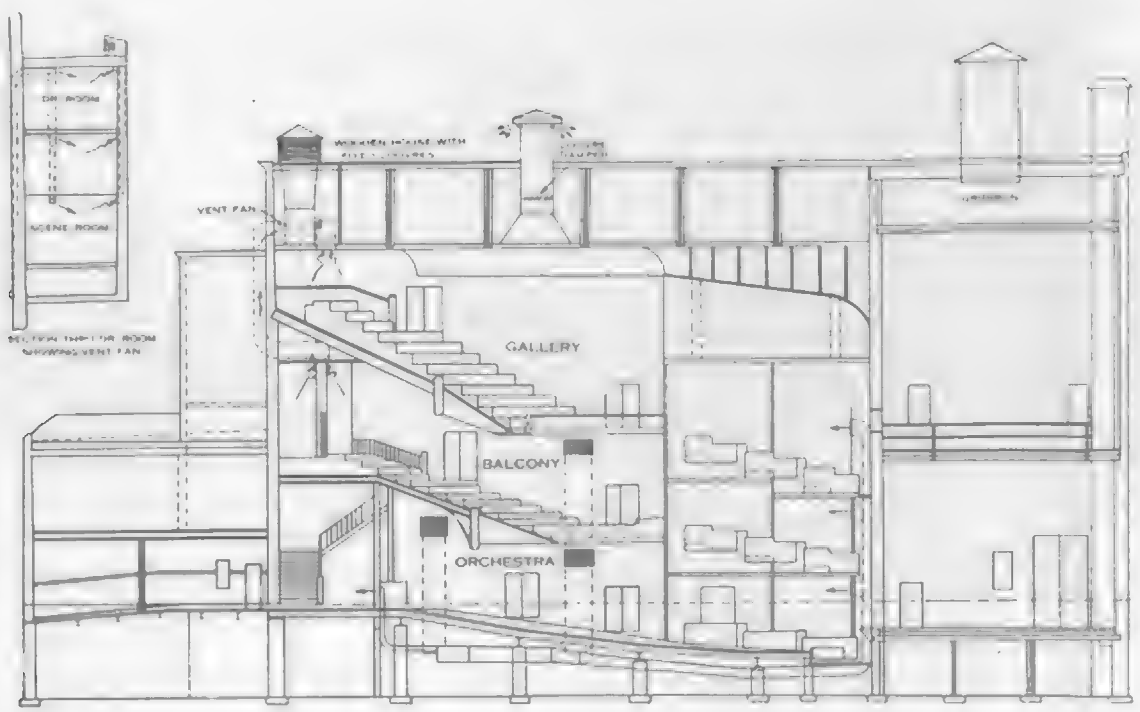
The modern school building offers the most exacting requirements in heating and ventilating. The large number of pupils seated in one room require a very rapid air change, and this must be accomplished without causing any drafts. A uniform temperature must be maintained throughout the room and the ventilation must be adequate.

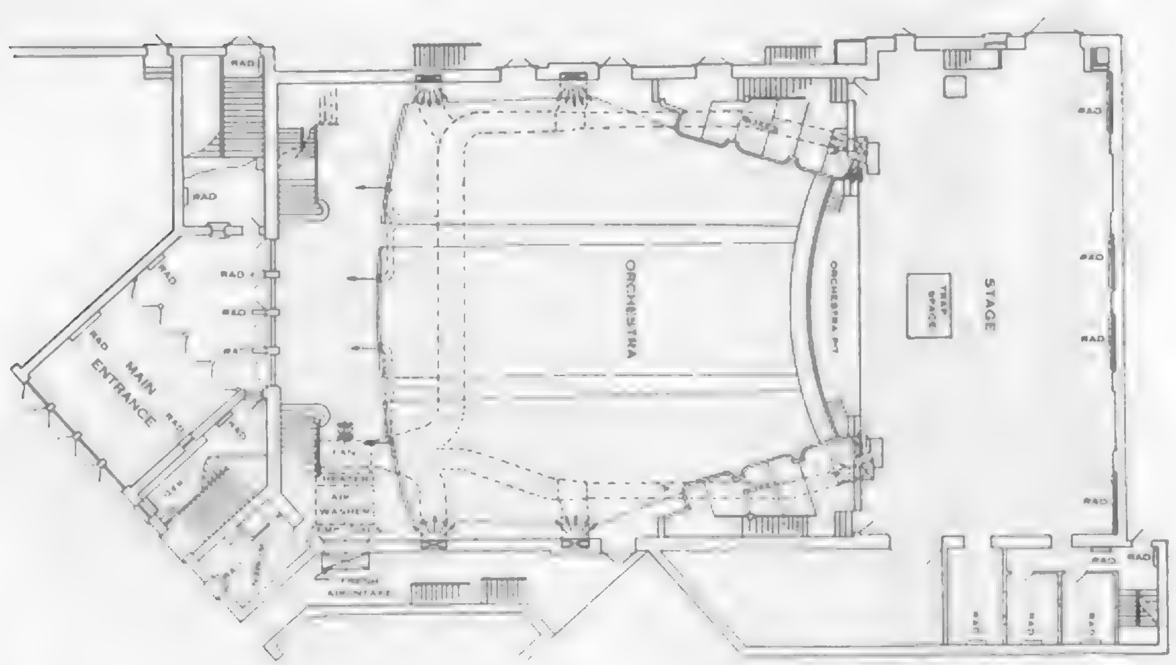
The Buffalo Fan system adapts itself very readily to the accomplishment of the former but the latter is somewhat more difficult to attain. Even elaborate system cannot secure an entirely perfect distribution of air and the only practical and successful method has been to supply air considerably in excess of the theoretical requirements. The necessity of this extra requirement or factor of safety as it might be called, is often overlooked in writing specifications for school ventilation.

Thirty cubic feet of air per pupil, which is the amount usually specified, will keep the CO_2 content down to from six to seven parts in 10,000. This supply would be ample if the air distribution were perfect but it has been found advisable that 40 cu. ft. per minute per pupil be introduced to insure the best results.

The Buffalo System has been installed in schools throughout the world with marked success.

The West Philadelphia High School is Buffalo Equipped





Theater Layout

Theaters and Churches

Audience halls, such as theaters, churches and lecture rooms though in use but for a short time are as a rule notorious for their poor ventilation. The introduction of the Buffalo Fan System has effectively relieved this disagreeable and unsanitary condition. Owing to the large dimensions of such buildings, and to the density to which they are peopled the problems of air distribution and avoidance of drafts are greatly increased.

Two plans have been found to give the best success in the ventilation of audience halls, these are usually distinguished as the upward and downward systems. In the downward system the air is admitted through registers in the walls at a height of several feet above the floor, and removed through vent registers in the walls at the floor line in the same manner as in school buildings or the air may be exhausted by means of separate disc fans placed in the walls of the building. In the upward method the air is admitted through duct outlets in the floor underneath the seats and is exhausted by means of disc fans in the walls or ventilators in the roof.

The upward method is to be desired wherever the architectural design makes it permissable. A perfect distribution of air can be secured, and the air flow is upward in accord with the natural air currents induced by the heat of the body and the breath. The products of respiration, and eliminations of the body are immediately carried away, and the incoming air is uncontaminated. This method of ventilation is exceedingly efficient, as a high standard of purity can be maintained at the breathing line with a comparatively small air supply. One objection to this system is that the air being introduced at the floor tends to carry up with it all loose dirt which may be raised from the floor by the action of people walking or moving their feet while seated.

With ordinary precautions as to the cleanliness of the floors this objection is for the most part overcome.

The moving picture theater has offered the largest field for theater ventilation in the last few years. The system most in vogue for these installations is the downward system. The air is introduced through registers in the side wall and exhausted by means of disc fans or ventilators. The ventilation requirements for audience halls and theaters are now very fully covered by legislation in most of the states.

Upward ventilation, to be successful, requires a very careful arrangement of the supply openings on account of the greater liability of drafts. The velocities are necessarily low, and the registers are so small that a very large number is needed to convey the necessary air.

The plenum chamber for supply is sometimes out of the question, and on this account the downward system, which is in almost universal use in schools, is extended to churches, theaters and halls with high ceilings. With a proper arrangement of fresh air and vent registers, and ample air supply excellent results are obtained. To insure such results exhaust systems are frequently relied upon,the vent registers being connected with suction fans which maintain a steady draft.

The design of theaters and churches often prevents the location of vent flues except in outside walls, the cooling effect of which seriously impairs the efficiency of the natural draft. It is always advisable to connect flues so located with suction fans.

In theaters which are in use during the summer, the air washer provides the means of securing freedom from distressing heat. In order to maintain the best cooling effect, refrigerating apparatus for lowering the temperature of the water sprays is sometimes necessary, and may be economically installed and operated, but even without the use of refrigerated water, the cooling effect is considerable and of decided practical value.

The air washer and cooling apparatus enables the temperature to be lowered about 10°, converting the theater from the most uncomfortable to the most comfortable place in warm weather, while in winter it gives a cleanliness and an increased freshness to the air supplied.

St. Paul's Cathedral
St. Paul, Minn.

Department Stores

Department stores offer an especially useful field for the application of the fan system. In cold weather there exist disagreeable cold drafts along the floors. Although on account of the crowded condition ventilation is most urgently needed, no provision as a rule is made for supplying it. The fan system fills both of these needs, first, by furnishing warmer air in large volumes without the production of drafts, and second, by creating an outward pressure which effectually prevents the entrance of cold air at the doors. The objection to the fan system previously existing on account of the dust carried into the building by the fan is entirely overcome in the Buffalo Fan System by the use of the air purifying apparatus, while at the same time the store is made very attractive in the hot days of summer by the effect which may be obtained when using this system of cooling.

The department stores shown below reap the benefits of Buffalo heating and ventilating systems. The Lord & Taylor installation admirably meets the varied weather conditions of New York City while, on the other hand, the delightful climate of Los Angeles is further enhanced by the Buffalo system in the Broadway Department Store.

THE BUFFALO FAN SYSTEM
OF
Heating, Ventilating and Humidifying

PART TWO

Industrial Plants

THE apparatus used in industrial building application is similar to that used in public buildings. The heating system ordinarily consists of three elements, namely; the heater, the fan and the system of air distributing ducts. In such installations where pure air and humidity control is required, an air washer is installed in addition to the equipment mentioned above. The draw through system is most commonly used in industrial work inasmuch as higher velocities are used than in public building application, and also due to the economy effected by the fan discharging directly into the duct system.

Heat Losses

In industrial buildings the heat losses are due to two causes; first, by the direct transmission of heat through the walls and exterior surfaces of the building, and second, by the infiltration of cold air from the outside. The loss due to the first cause may be calculated very closely in accordance with the method described on page 74, but the heat loss due to infiltration differs so greatly in various sizes and construction of buildings that no definite rule can be laid down. The allowance to be made for this is necessarily the result of experience and of careful tests of previous installations. The most effective remedy to reduce this loss to a minimum is to maintain a slight pressure or plenum within the building by means of a fan.

Fan System vs. Direct Radiation

In all heating systems difficulty is experienced due to the rise of heated air before its heat has been utilized to the fullest extent. This heated air forms a stratum just beneath the roof. In the modern type of factory construction with its height and great amount of skylight surface, the loss due to this action of the heated air may be considerable and its prevention is a serious problem. In direct radiation where the air current is entirely due to the difference in temperature, the attendant loss, which is relatively great, is unavoidable. Practically, the only way in which this heated air can be made use of is by placing the coils next to the wall near the floor, and allowing the heated current of air to pass upward along the walls, but this method is extremely wasteful, due to the fact that part of the heat is applied directly to the walls, causing a loss estimated as great as 25% of the total heat supplied.

With the fan system on the other hand, the method of distributing the air is entirely mechanical, and thus an opportunity is afforded for utilizing its heating effect to the very best advantage. The method of distribution may be so devised that the effect of a rising current of heated air is almost entirely avoided, this being secured by diffusion of the heated air along or near the floor line.

The Buffalo Fan System possesses a great advantage over direct radiation systems in its flexibility of operation. With direct radiation a building heats up very slowly, and it is usually necessary to maintain a normal temperature all night in order to have it sufficiently warm in the morning. On the other hand the fan system with the proper amount of reserve can heat a building up in a short time. This allows the building to be cooled down during the night to just above freezing point, say an average temperature of 35° or 40° where the manufacturing process will permit.

Another important point of economy in the Buffalo Fan System is the utilization of waste sources of heat. The most common form of waste heat in an industrial building is from steam engines and other steam driven machinery.

The ordinary simple steam engine running non-condensing has a water rate of about 32 pounds per horse power. Of the total heat supplied by the steam only about 20% is utilized in work leaving 80% of the heat unused. A great portion of this remaining 80% is available for use in heating apparatus, a small part being lost due to radiation.

Since the mean effective pressure in an ordinary engine cylinder may be placed at 40 pounds per square inch it will be seen that an increase of one pound per square inch in back pressure will reduce the effective horse power of the engine two and one-half per cent. and correspondingly increase the cost of the power produced.

In a compound engine the effect of back pressure is still more noticeable since the mean effective pressure, referred to the low pressure cylinder, may be placed at 30 pounds per square inch; each pound of back pressure therefore reducing the power of the engine three and one-third per cent. It is therefore unprofitable to introduce any system that will greatly increase the back pressure on the engines.

The ordinary system of direct radiation places a back pressure on the engines which is prohibitive. On the other hand the Buffalo Fan System Heater is designed for use of steam at low pressure and can be operated successfully with one-half pound back pressure on the engine.

Heating with Exhaust Steam

The question is frequently brought up whether it is cheaper to run an engine non-condensing and use exhaust steam for heating or to run the engines condensing and use live steam for heating purposes. With the average compound Corliss engine the water rate at full load is about 20 pounds per horse power when running non-condensing and about 14 pounds condensing, so that a saving of 30% in the water rate is effected when running condensing.

The amount of heat available in the exhaust steam is about 80% of the total. Hence it will be seen that the saving of steam when running condensing is only six pounds per horse power, while the heat available in the exhaust steam is equivalent to 16 pounds of steam per horse power and therefore a saving of the equivalent ten pounds of steam per horse power could be saved by running the engine non-condensing and using the exhaust steam in the heater. In this manner a saving is

effected as long as 38% of the steam is utilized in the heater. With engines whose economy is less than that assumed above, the saving effected by running non-condensing and using the exhaust steam in the heaters is even greater.

With the steam turbine the water rate increases much more rapidly with a decrease in vacuum than in the case of the steam engine. A steam turbine having a water rate of 20 pounds of steam per horse power with 28 inches of vacuum will require 50 pounds of steam per horse power when running non-condensing. From this it is readily seen that the use of exhaust steam from a turbine running non-condensing is economical when the heating requirements are more than 60% of the steam consumption of the turbine when running non-condensing.

Besides these distinct advantages in economy over direct radiation there is usually a considerable advantage in first cost in favor of the Buffalo Fan System. This is due in part to the compactness of the system, requiring fewer connections and shorter lengths of steam mains, but more particularly to the great saving in amount of radiating surface required owing to its greater effectiveness in the fan system. A determining factor in the rate of heat transmission of any heating surface is the velocity of air over the surface. This is shown by the curve on page 72, exhibiting the relation between air velocities and heat transmission as determined by accurate tests on the Buffalo Fan System heater. In direct radiation the heat is transmitted by convection currents and radiation only, while with the fan system an air velocity over the coils of from 1,000 to 1,200 feet per minute is usual; the former transmits only from 2 to 2.6 British Thermal Units per square foot per hour, per degree difference in temperature, while the fan system heater as shown by the curve on page 73, transmits from 10.4 to 11.5 B. t. u. per square foot per hour, per degree difference in temperature or about five times as much as direct radiation. Hence a correspondingly smaller amount of radiating surface may be used, which more than offsets the additional cost of fan, engine, and hot air piping.

The question often arises as to the relative cost of heating, ventilating, and humidifying. As an example, assume a fan system of heating in a schoolroom, where outside temperature is 0° and room temperature is to be kept at 70°. Air must be raised to 70° before any heating will be done by it, therefore consider this amount of heat added for ventilation purposes.

The temperature of the air has to be raised still further for heating the room, and it is ordinarily assumed that air entering a room at 120° with outside temperature 0°, will probably take care of heating requirements, and also furnish a sufficiently rapid air change.

Accordingly 70° of 120° total or 58%, is used for ventilation and 42% for heating; and approximately the cost of ventilation is 60% and the cost of heating 40%, where humidifying is not considered.

Assuming that this same proportion holds for other temperatures, when the outside air is 40° and the room is to be kept at 70°, 30° or 58% is the amount of heating required for ventilation, and 22° or 42% for heating; and temperature of air entering the room should be 92°.

The amount of moisture which air will contain depends on its temperature. The amount of moisture actually contained at any temperature is called the absolute humidity; and the ratio of moisture which air actually contains at any temperature compared to what it could hold at that same temperature, is called the relative humidity. Thus, if a cubic foot of air contains 0.5 gr. of moisture at 0°,

this being its absolute humidity, the absolute humidity will be 0.5 gr. when the air is heated to 70°. But a cubic foot of air at 70° would be capable of containing 8.0 gr. of moisture, therefore its relative humidity at 70° would be only about six per cent.

When outside air is about 30°, it is well to have about 4 to 5.5 gr. of moisture per cubic foot of air, when temperature is raised to 70°; but with outside temperature 0° it is ordinarily considered that relative humidity should be about one-half the difference between outside and indoor temperatures, or where outside air is 0° and room temperature 70°, the relative humidity should be about 35%. This is the practical value which will not cause steaming of windows.

Assume in the above example that 35% relative humidity at 70° is to be maintained. The air then would leave the humidifier completely saturated at 41°, containing 2.85 gr. of moisture per cubic foot, and then could be raised to any desired temperature by passing over heating coils. As air entered at 0° containing 0.5 gr. of moisture per cubic foot, 2.35 gr. of moisture should be added to each cubic foot of air. Through the ordinary range of temperatures the absorption of one grain of moisture per cubic foot lowers the dry bulb temperature 8.5°, or 8.5° are necessary to raise moisture in a cubic foot of air one grain or 20° will be necessary to raise moisture per cubic foot 2.35 grains.

This will be in addition to the 120° for heating and ventilating, or 140° will be required for heating, ventilating, and humidifying. Therefore 70° of 140° total, or 50%, is required for ventilating, 36% for heating, and 14% for humidifying, and it can be stated approximately that cost of ventilating will be 50%, cost of heating 35%, and the cost of humidifying 15%.

Systems of Air Supply

The method of distributing the air in an industrial building is a consideration of chief importance. The methods usually applied are as follows:

First, the air is taken entirely from without and after being heated is forced directly into the building through the distributing ducts, this method being generally known as the Plenum System. The pressure produced within the building causes continuous exit of the air from the building, either through the natural openings, as is usually the case in factories and other large buildings, or through special vent openings provided for the purpose as described under Public Building Application. This method effectually prevents the entrance of cold air from without.

A second and by far the most common method used in industrial plants is to draw the supply of air entirely from within the building, raise it to the proper temperature and force it through the distributing ducts thus causing a continuous circulation within the building itself. This method can be used in industrial applications since the question of ventilation is not of as great importance as in public buildings, inasmuch as the relative amount of air per occupant is very much greater in industrial plants.

The ideal arrangement is a combination of the two mentioned above and should be used wherever possible. In this method, the greater portion of the air is returned to the apparatus, but sufficient fresh air is taken from the outside to create a plenum within the building and thus prevent the inward leakage of cold air. In this manner the amount of air loss by leakage is made up—not by the infiltration of cold air through the crevices around the doors and windows—but by air that has passed through the apparatus and has been heated to an effective

degree. This combination has been found to be more economical than where all returned air is used. The proper amount of air to be taken from outside is determined by securing a condition within the building so that the noticeably inward flow of air around the doors or windows ceases. If the plenum is carried beyond this point, there will be a loss due to the heating of an excess amount of outside air drawn through the apparatus.

Systems of Air Distribution

The vertical duct system such as usually used in public and office buildings is frequently employed in factory buildings. In this system the air is admitted through vertical ducts or flues built in the walls and opening into the room at a point about eight feet above the floor line; suitable openings being supplied at the floor line connecting either with vents opening to the roof, or an exhaust duct system through which the air is drawn out. By this method the heated air is continuously forced downward as it cools, and the cool air is removed at the floor line.

This system may be modified by placing the ducts in the room along the walls and either blowing the air out at a height of about eight feet or very close to the floor and blowing directly downward along the floor. The latter method secures a perfect diffusion of the heated air at the floor line and avoids any draft which would be objectionable. In buildings having a large open area a system of overhead piping is installed to the best advantage. Excellent results are obtained by this method providing the pipes are not placed at too great a distance from the floor. The chief advantage of the overhead system is a saving of initial cost, since on account of the high temperature and velocity of air in the distributing pipes, a great amount of heat can be transferred with a very small amount of material necessary, thus the cost of the galvanized iron distributing system air ducts is relatively small. The best results are secured with outlets at from 12 to 18 feet above the floor line. When running the ducts at this height the air may be blown out directly by means of short connections. Above this height it is preferable to use drop pipes extending downward along the structural columns so that they will not interfere with any moving mechanism.

Another system which has proven very satisfactory is that in which a distributing air return duct is employed. This approaches in principle very close to the plenum system as used in public buildings and is a combination of both the plenum and exhaust systems. In this system no distributing ducts or piping for the heated air are used but small fan units are placed at intervals throughout the building. The air is blown directly into the building at about eight or ten feet above the floor through an outlet coming directly from the fan and having short outlets branching in several directions. Return vent ducts are placed at frequent intervals along the wall, these leading into large return air tunnels or ducts through which the air is drawn by the heating fans. In this method the circulation is effected entirely by the return vent ducts rather than by the hot air ducts. This method is to be recommended where an elaborate duct distributing system is impracticable or undesirable.

This system has marked advantage over all other systems in that piping cost is cut to a minimum due to the high velocities and high temperature of air handled by the fan and that a positive circulation of air is produced.

Industrial Applications

The world is progressing and working conditions of years ago are no longer tolerated. The progress in machine tool design and increased production has bettered the class of workmen. The former artisan is now a specialist.

It is a recognized fact that atmospheric conditions have a marked effect upon the comfort and efficiency of a workman. Thus the maintenance of proper atmospheric conditions within a plant pays big returns in comfort and contentment of the workmen themselves and in increased and better production.

The Buffalo Fan System of Heating and Ventilating is in successful operation in every type of factory building and in connection with every form of industry. A mere list would take up more space than advisable in this volume so we will content ourselves with the recital of just a few applications as given below.

Machine Shops

The requirements of the modern machine shop are most admirably met by the Buffalo Fan System of Heating and Ventilating. The modern construction with its large areas of glass roof and large single room volumes presents a very perplexing problem in both air distribution and pressure balancing. How successful our engineers have been can best be shown by describing the system as in operation at the National Acme plant at Cleveland, Ohio.

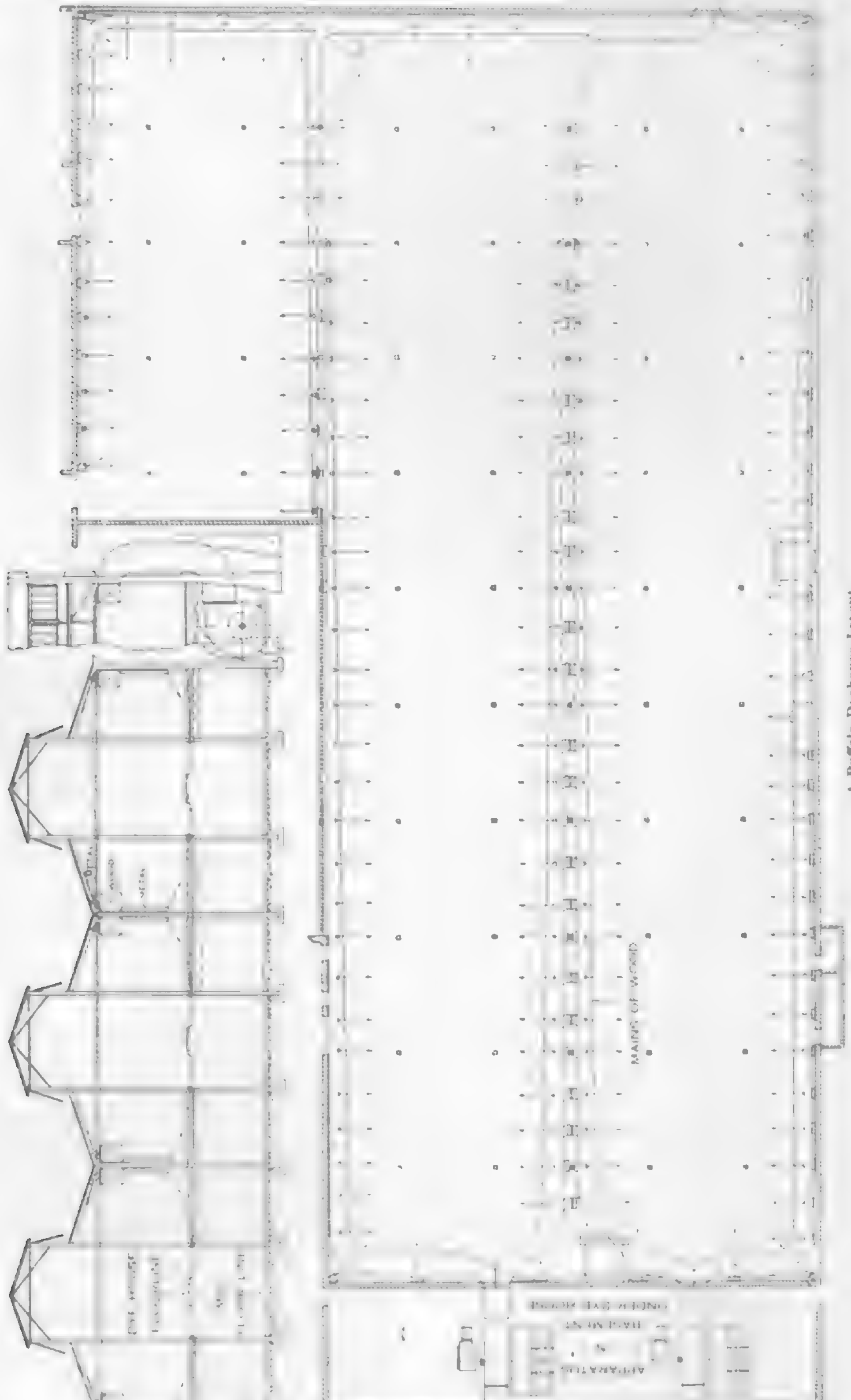
This plant consists of a one story building of saw tooth construction and covers seven and three-fourths acres. This ranks as one of the largest machine shops in the world. The whole floor is covered very compactly with automatic machines for making bolts and nuts.

The heat losses from the side walls of brick and glass are taken care of by direct radiation and the Buffalo System takes care of the other heat losses which are by far the greater portion.

There are four sets of apparatus each consisting of an exhaust fan returning air from the floor line and discharging it either into the inlet of the air washer for supply or into the atmosphere through ducts through the roof, a supply fan taking air from the exhaust fan or from outdoors as the conditions require, an air washer and the heater units. Both fans are driven by silent chains from a 12″x14″ engine. The exhaust fans are provided with an auxiliary motor drive so they can be driven independently when the supply fans are not in operation. Each unit handles 28,000 cubic feet of air per minute and has 6,720 square feet of heater surface.

The fresh air inlet dampers and the exhaust fan discharge dampers are automatically controlled by a thermostat located in the discharge of each air washer.

A considerable quantity of oil vapor and fumes are given off by the automatic machines and the apparatus when in operation keeps the building remarkably free from any traces of these.



A Buffalo Dyehouse Layout

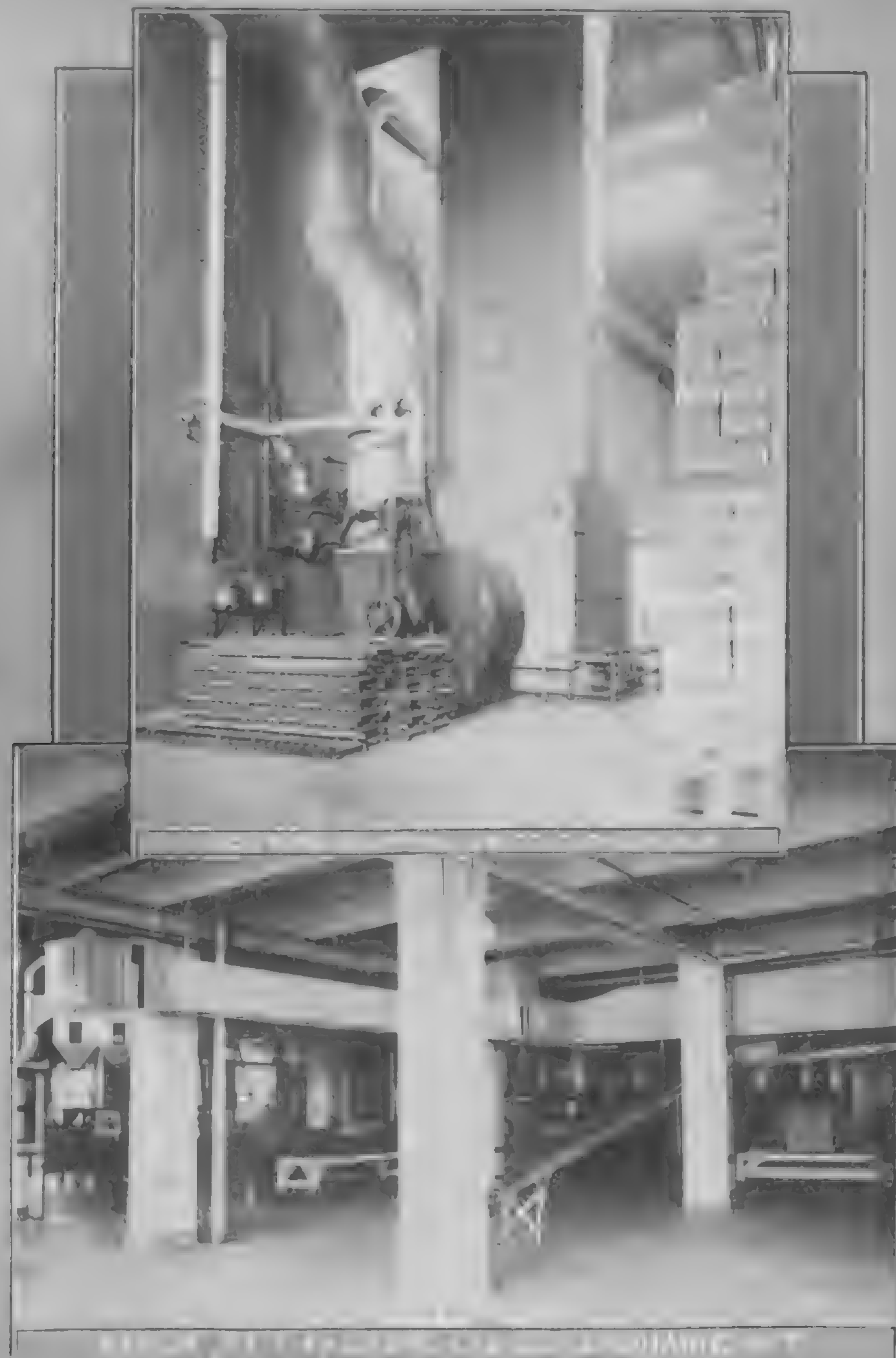
Dye Houses

The dye house presents a problem in ventilation that is peculiar and distinctive. The large amount of steam present has always caused trouble by condensation of all cool surfaces throughout the building, making a most undesirable atmosphere to work in and also causing excessive deterioration of the building itself.

Our engineers have made a successful study of this problem and the introduction of the Buffalo Fan System has made the dye house so equipped just as livable as any other part of the factory and removed all traces of condensation on the interior of the building. The entire secret of successful dye house installation is to apply the correct amount of air at the right place.

This is accomplished by blowing heated air into the room just above the dye vats and machines and blowing a current of heated air along the surface upon which the vapor has a tendency to condense.

The air blown across the vats and machines dissipates the steam and the other forms a current or film of heated air along the cool surface so that the moisture-laden air is insulated from these cool surfaces. The air is removed by means of ventilators in the roof or disc fans placed at various points in the walls or by a combination of both ventilators and fans. By this method a rapid absorption and removal of all moisture is effected.



The dye house of the Pacific Mills at Lawrence, Massachusetts is Buffalo equipped. This dye house ranks with the largest in the country and is absolutely free from steam and condensation due to the efficiency of the Buffalo System.

The apparatus consists of one No. 12 double width Turbo Conoidal fan, two No. 16 Niagara fans and one No. 19 Niagara fan, sixteen 48" propellor fans and 11,000 square feet of Heaters.

Paper Mills

Paper Mills present one of the most fertile fields for air heating, ventilating and humidifying.

In the machine room we have the cold damp portion around the wet end of the machine, the hot humid portion around the driers and the cool portion around the calendar end. The center of the room is over-heated while the ends require additional heat to make them livable. Along with these we have the constant dripping of condensed moisture from the roof. This condensate drops down upon the paper and causes great loss by injuring the finished product in addition to the rapid depreciation of the roof construction. The grinder rooms are extremely cold and damp and the roof condensation is also quite a problem.

The problem met in paper mills is similar to that in dye houses. Warm air must be introduced into the room without conflicting with the natural air current tendencies and ample provision must be made for exhausting the moisture-laden air without allowing it to come in contact with the cool surfaces of the building.

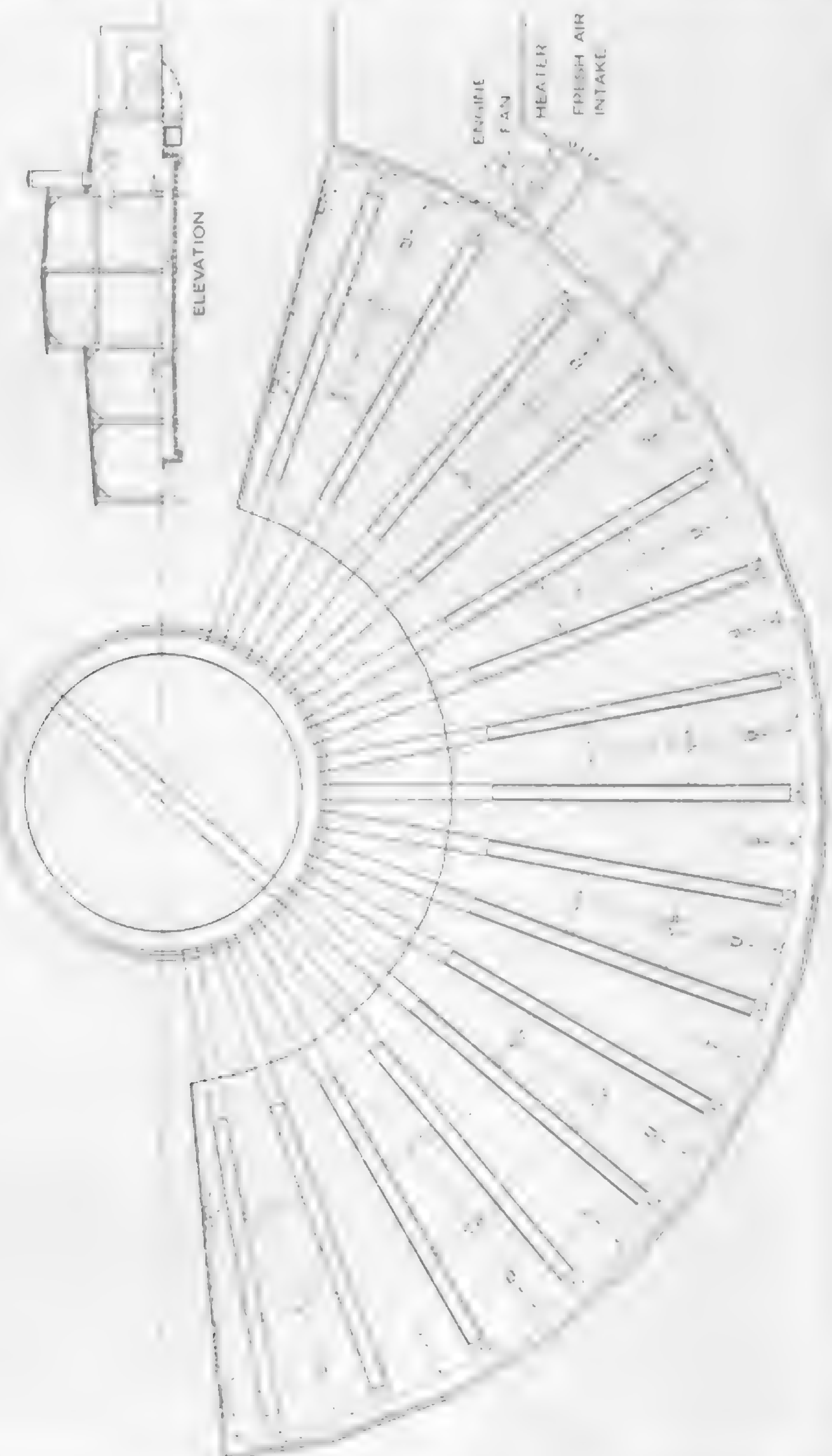
The S. D. Warren Co., Cumberland Mills, Maine, and the Eastern Manufacturing Co., at Brewer, Maine, head the list of paper mills reaping the benefits from Buffalo Heating and Ventilating.

Food Product Plants

In plants where food products are handled, the chief requisite of the heating and ventilating apparatus is that the air delivered to the workrooms be absolutely clean. In addition to this a uniform atmospheric condition must be maintained, for it has been found that the quality of the product changes with variations in the atmospheric conditions under which they are prepared. Both of these requirements are admirably met with the Buffalo system of heating and ventilating equipped with a Carrier air washer. The effectiveness of the Carrier air washer is shown by the picture on page 17.

The well known products of the Beech Nut Packing Company at Canajoharie, New York, are all prepared and packed in the presence of pure, clean air delivered by Buffalo Apparatus. Not only is the air washed and heated but is also delivered to each room at the exact humidity required for the process taking place in the room.

Campbell Soups are also benefited by Buffalo heating and ventilating.



A Typical Round House Layout

Railroad Round Houses

Round houses present a very difficult heating problem due to the large volume of warm air carried off through the open smoke jacks which act as ventilators. A great amount of heat is absorbed, too, in the melting of the snow and ice on the locomotives and in the evaporation of the moisture thus produced. Ample ventilation is required to carry off the smoke and gases and considerable heat is required due to the excessive ventilation requirements.

The usual method employed is to draw the air direct from outside and after passing through the coils of the heaters to distribute the air by means of underground ducts discharging into the pits directly under the engines. The outlets are often fitted with volume regulating dampers.

This is very clearly shown in the drawing on the opposite page. In addition to the outlets in the pits the cold outside walls are taken care of by outlets along some of the columns and blowing toward the cold walls.

The cut below shows the Buffalo Fan used for heating and ventilating the N. Y. C. R. R. round house at Gardenville, N. Y.

Advantages of the Fan System

The chief points of superiority of the Buffalo Fan System may be summarized as follows:

1. Perfect ventilation regardless of exterior conditions.
2. Uniform and proper distribution of heat.
3. High efficiency of heating surface (three to five times that of direct radiation).
4. Greatest economy in operation.
5. Utilization of exhaust steam.
6. Prevention of cold drafts from without by production of a plenum.
7. Independent regulation of heating and ventilating effects.
8. Great flexibility in operation to suit varying conditions, affording a maximum economy.
9. Ease of control, which prevents over-heating.
10. Great compactness, affording an economy of space and reducing the cost of steam connections.
11. Perfect drainage, making less repairs necessary and giving a lower rate of deterioration than with direct radiation.
12. Low cost of installation.
13. The entire apparatus is easily portable and is, therefore, a permanent asset.

THE BUFFALO FAN SYSTEM

OF

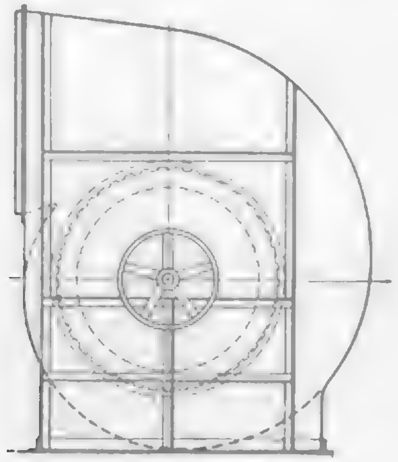
Heating, Ventilating and Humidifying

PART THREE

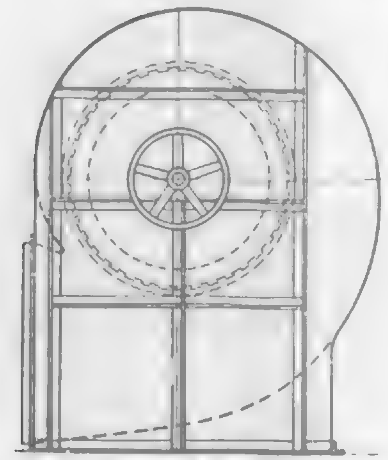
Buffalo Apparatus

THE Buffalo Fan System Apparatus consists of a fan, an engine or motor, some form of indirect heating coil, and a Carrier Air Washer and Humidifier. The general arrangement may be either the EXHAUST or DRAW THROUGH system in which the air passes through the heater before reaching the fan, or the BLOW THROUGH in which the fan is in front of the heater and blows the air through the heater coils. The selection of the arrangement to be used depends upon the individual requirements of the location, each arrangement having its own peculiar advantages. The exhaust through apparatus possesses the advantage of greater compactness and a more convenient arrangement. On the other hand, the blow through apparatus is larger but occupies a more narrow space. The former requires the use of an exhaust fan, one having only one inlet, which is slightly less effective than a blower having two inlets such as is used in the blow through type; however, the exhauster discharges directly into the duct system without any reduction in the velocities of the heated air so that all the energy of velocity of discharge is utilized. The blow through system on the other hand requires a change from the relatively high velocity at the fan outlet to a low velocity through the heater and back again to a high velocity upon entering the air ducts which causes an unavoidable loss in pressure.

Due to its compactness the exhaust through apparatus is customarily employed in factory buildings. The blow through apparatus is necessarily used in public buildings and elsewhere wherever independent temperature regulation is demanded as the use of a by-pass around the heater permits the independent distribution of hot air and tempered air in any desired proportions.



1 Full Housing
Top Horizontal



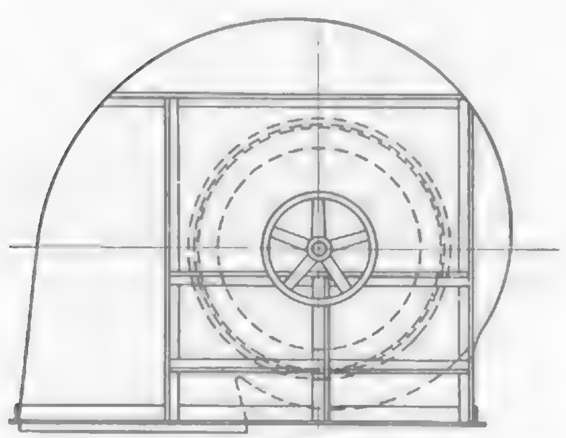
2 Full Housing
Bottom Horizontal

Fans

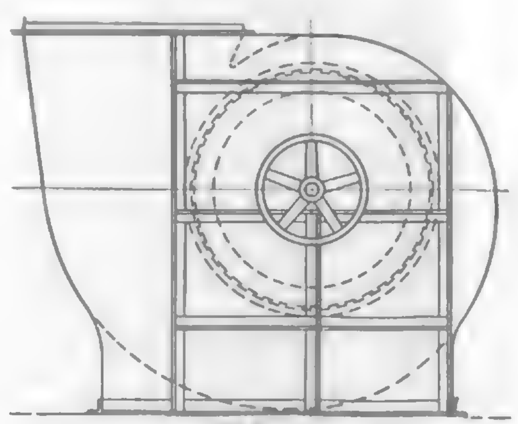
Fans and blowers are designated by the position of the discharge opening and are classified as follows:

Top or bottom horizontal discharge, up or down blast, and special, the latter being described by giving the angle of discharge from the horizontal. The hand of a fan or blower is determined by the side on which the pulley or engine is located. When facing the discharge outlet, the fan is either left or right hand according to whether the pulley is on the left or right side as seen from this position.

A brief description of the various types of fans manufactured by the Buffalo Forge Company follows.



3 Full Housing
Down Discharge



Full Housing
Up Discharge

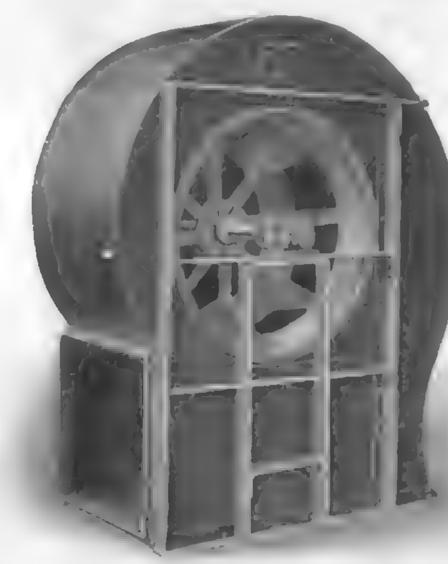
Buffalo Cone Fan

Buffalo Planoidal Fan Wheel

Cone Fans

For the introducing of cooled or tempered air into rooms where no distributing system is required or for exhaust ventilation where the resistance to be overcome is moderate a type of fan known as the cone wheel is suitable. This special form of fan wheel is used without a housing and is shown in the cut above. This fan wheel must not, however, be compared with the disk or propeller fan, since it is purely of the centrifugal type. Tables of performance are found on page 76.

Planoidal (Type L)



Buffalo Planoidal Fan Type L
Full Housing Bottom Horizontal Discharge

One of the first developments of the centrifugal type of fan wheel was the steel plate fan. In this fan the blades consist of flat radial plates and are few in number. As the result of extensive experimenting and testing by our engineers the Planoidal (Type L) steel plate fan was developed which was a distinct improvement over the old style steel plate fan. This fan is provided with an inlet cone on the housing and the proportions of the wheel, housing, inlets and outlets were so designed as to materially increase the capacity and efficiency, at the same time reducing the power consumption. Tables giving the ratings are found on pages 78 and 106.

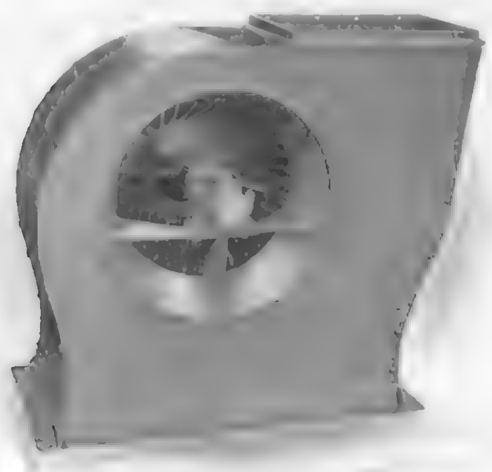
Niagara Conoidal (Type N) Fans

With the increase in the speed of prime movers it was found necessary to design fans to operate at a higher speed and one of the marked developments in this line was the Buffalo Niagara Conoidal Multiblade fan. This fan derives its name from the prevalence of conical shapes in its design. The blades are made to conform to the tapering surface of a cone, the inlet is conical and the blast wheel forms the frustrum of a cone.

These characteristics are very clearly shown in the adjacent cut.

Fans from No. 3 to No. 6 in size are made with cast iron inlet bearing stand and cone as shown below. All sizes over No. 6 are made with sheet iron inlet cone and flat steel bearing standards as shown in cut below.

Performance data will be found on pages 79 and 107.

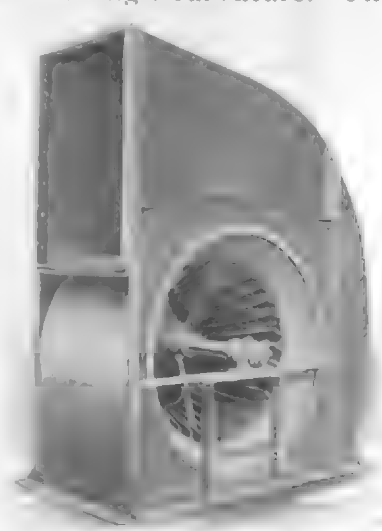


No. 3 to No. 6 Niagara Conoidal Fan, Right-Hand Up Discharge

Turbo Conoidal Fans

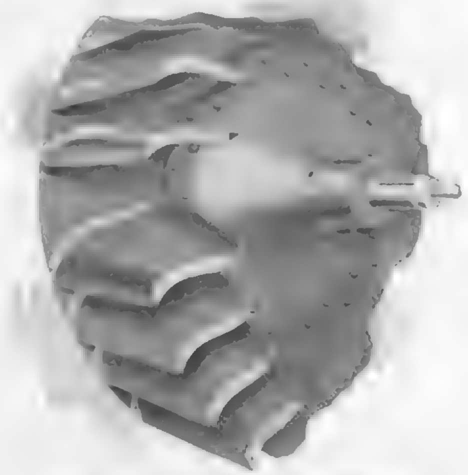
The increasing demand for air at high pressure was forseen by the Buffalo Forge Company, and a new type of Multiblade fan known as the Turbo Conoidal was developed. This fan differs from the Niagara Conoidal, only in that its blades are of double curvature instead of single curvature. This fan is particularly suited for operation where both high speeds and high pressures are essential. The various points considered in the design of the Niagara Conoidal fan were also taken into account by our engineers in the design of the Turbo Conoidal, and all parts are co-ordinated with the view of obtaining the highest efficiency with the lowest power consumption. Performance tables are found on pages 80 and 108.

All parts of the Niagara and Turbo Conoidal fans have been designed with the view of obtaining the best efficiency under practical operating conditions.



Three-Quarter Housing Niagara Conoidal Fan, Left-Hand Top Horizontal Discharge

The wheels, blades and hub are designed so that the air shall have a smooth easy flow from inlet to outlet without any abrupt change of direction at any point; also, the width of the blade is so proportioned that the back part cannot take up any greater part of the air, this prevents uneven pressure and eddy currents, and effects an even distribution of the air over the entire surface of the blade. Our success in this design has been proven by practical tests, and our standard guarantee is that the velocity of air issuing from any part of the fan outlet as measured by a Pitot tube is not more than 15% above or below the average velocity across the entire opening.



Turbo Conoidal Fan Wheel Partly Assembled, Showing Double Curvature of Blades

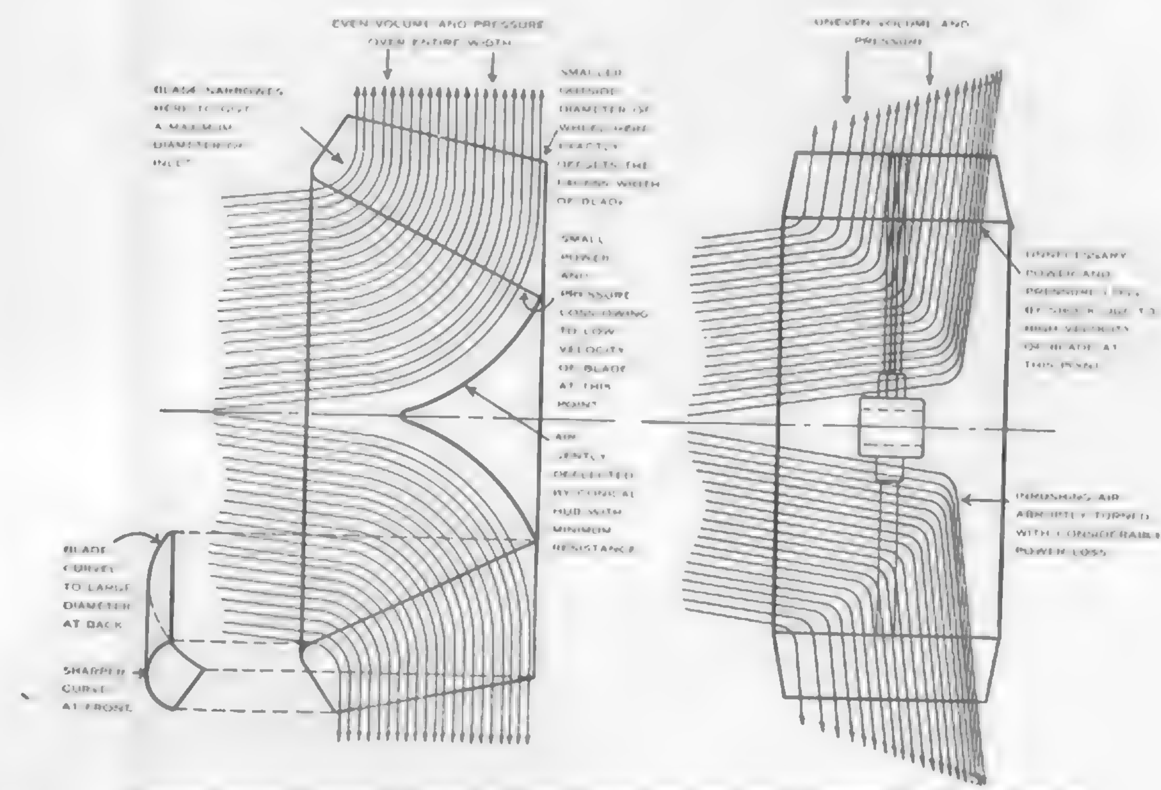
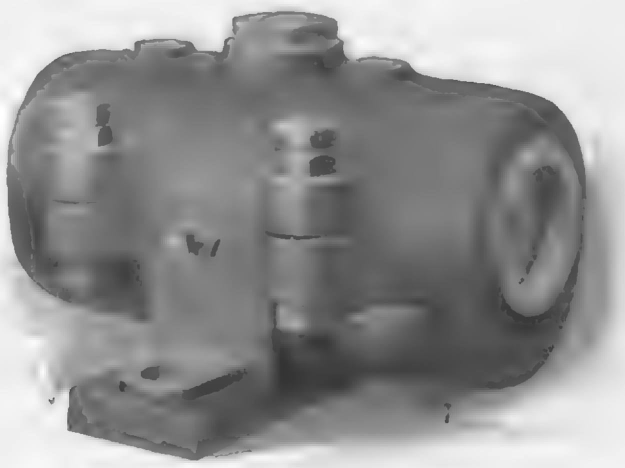


Diagram Showing Advantages of Niagara and Turbo Conoidal over other types of Multiblade Fans in Handling Air

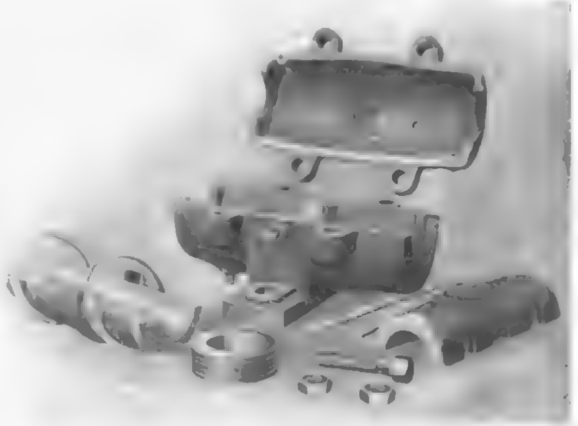


Buffalo Spherical Type Fan Bearing

One of the prominent features of Buffalo Fan construction is the type of bearing used. It was proven early in the history of fan construction that the reliability of operation of a fan was in a large measure determined by serviceability of the bearing used.

The type of bearing described below is by far the best fan bearing on the market today.

This dust proof and oil tight bearing consists of a split sleeve lined with babbitt and completely encased in the bearing housing. The sleeve is mounted between spherical surfaces which allows the bearing to adjust itself in every direction, and the housing provides a large oil reservoir in which two oil rings dip; overfilling is prevented by the position of the opening through which the oil is supplied and which also indicates the oil level.



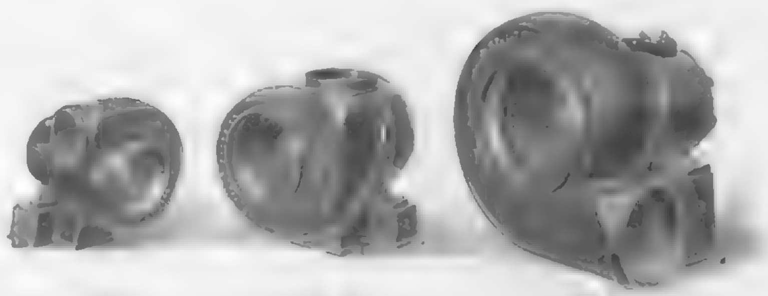
In the interest of safety the thrust collar is placed inside the housing, running against a babbitted shoulder; grooves on the outside surface of the thrust collar throw off all oil and absolutely prevent it from creeping along the shaft and being drawn into the fan.

Selection of a Fan

It has been proven both in theory and practice that the length of blade in a straight blade fan wheel is the deciding factor in the pressure obtained at a fixed rotative speed and that a curvature of the blade in the direction of rotation tends to increase the pressure. Whereas the curvature against the direction of rotation tends to decrease the pressure. It is often stated that the forward curvature of fan blades will increase the efficiency over that obtained with radial blades or backward curvature blades, this however is not true. Each type is admirably suited for a certain purpose; It has been found that short curved blades require a greater number for good efficiency than blades of the radial type, similar to the steel plate and Type L fans. With the steel plate or planoidal fan having a small number of radial blades usually from five to twelve depending upon the size, the pressure tends to build up as the capacity falls off, that is, at a constant speed the pressure is greater at half capacity than at normal rating. With the multiblade type, such as the Niagara Conoidal having single curvature blades, the pressure is developed more by change of direction and impact of the blades against the air, rather than by centrifugal force, the pressure is greatest at the normal load for which the fan was designed and decreased for any load, either above or below this normal capacity. This feature has been overcome in the fans having double curvature blades, e. g., the Turbo Conoidal in which the pressure builds up as the capacity falls off, in this respect being very similar to the steel plate fans. These points are very clearly brought out in the characteristic curves of these various types of fans as shown on pages 81 to 83.

From this it will be seen that in systems where a uniform air quantity is desired, whether for heating, ventilating, forced drafts or drying processes the steel plate fan and turbo conoidal fan will come nearer giving this uniform quantity in spite of variations in resistance brought about by throttling of dampers or similar conditions. On the other hand, it is sometimes very desirable to be able to cut down the capacity of a fan without increasing the pressure and velocity, as for instance, if one part of a building should be shut off; in a case such as this, the steel plate fan would deliver an increased amount of air into the remaining part of the system on account of the increased pressure, whereas the multi-blade fan of the Niagara type would be more sensitive to the increased resistance and would fall off in capacity due to this. In general multi-blade fans of equal capacity and efficiency require less space than steel plate fans and have the advantage of operating at higher speeds.

When specifying a multi-blade fan with single curvature blades extreme precaution must be taken in designing the duct system, in determining the frictional resistances of the entire system, and selecting the proper speed for the size of fan to be used.



No. 1 No. 2 No. 3

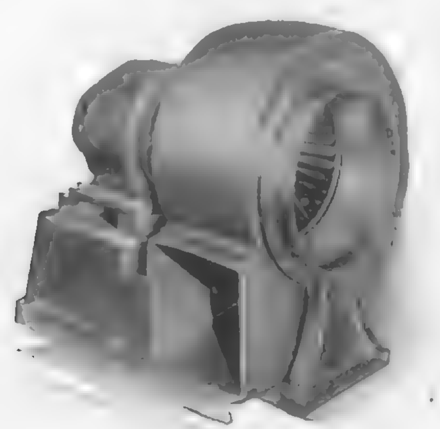
Buffalo Baby Conoidal Fans

The Baby Conoidal fan is of the high efficiency multiblade type with blast wheel of the same design as the Niagara Conoidal (Type N) which has met with such marked success. Housing is cast iron and can be swung around to discharge in any desired direction. This fan furnishes a large volume of air at a relatively low pressure with moderate speed. The wheel is accurately balanced, assuring a smooth-running, noiseless machine.

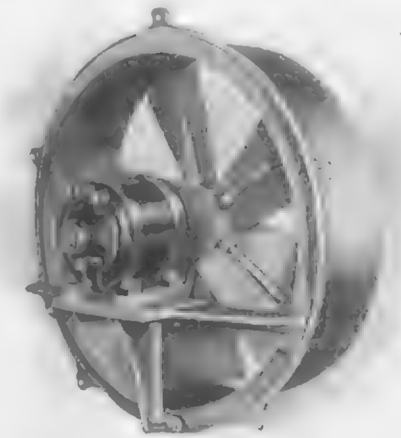
It is unexcelled for all kinds of drying and cooling purposes, for supplying fresh, cool air to offices, homes, staterooms, telephone booths, etc., and for exhausting smoke, fumes and foul air from kitchens, restaurants, lavatories, etc.

Cord and plug are furnished with No. 3 and smaller; no expense for installing, simply attach to an electric light socket. Outfits are furnished with 110 or 220 Volt D. C. motors and 110 or 220 Volt single phase, 60 cycle, A. C. motors. Nos. 4, 5 and 6 are also furnished with 110 or 220 Volt, two or three phase, 60 cycle motors.

Tables of dimensions and performance on page 76.



No. 6. Baby Conoidal Fan



Motor Driven (Type D)

Pulley Driven (Type D)

Disc Fans (Type D)

The ordinary disk or propeller fans are designed for use where low pressure heads are operated against. This type of fan should never be used in connection with a pipe system but should discharge directly into a room, or exhaust from it without obstruction. The characteristics of the Type D fan are very clearly shown in the above cuts.

Performance tables are given on page 77.

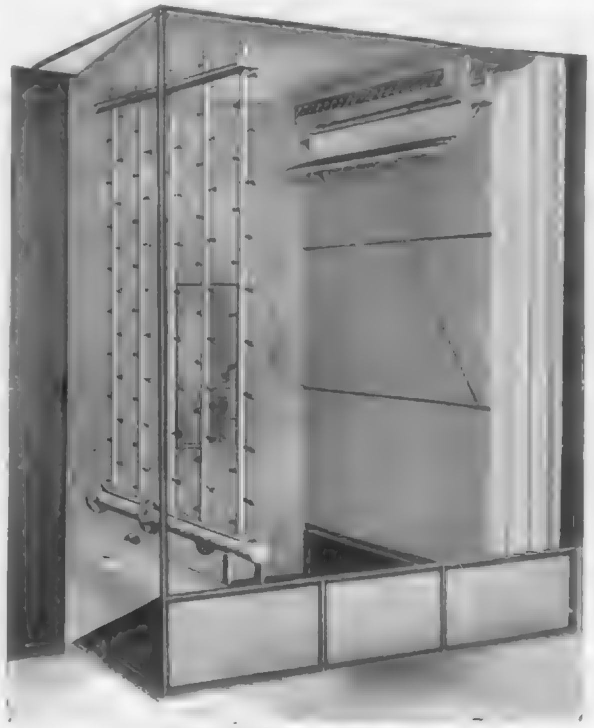
Disc Fan (Type CM)

Where a disc fan is used to overcome a moderate resistance, the Type CM with overlapping blades is recommended. This type of fan is used as a booster in mine ventilation, or for producing air flow in cooling towers for condensing plants. The casing and bearings are self contained.

Type CM

The Carrier Air Washer and Humidifier

The Carrier Air Washer consists of a spray chamber, a series of spray nozzles and eliminator plates. The air is drawn through the spray chamber where it comes in intimate contact with an atomized spray of water.



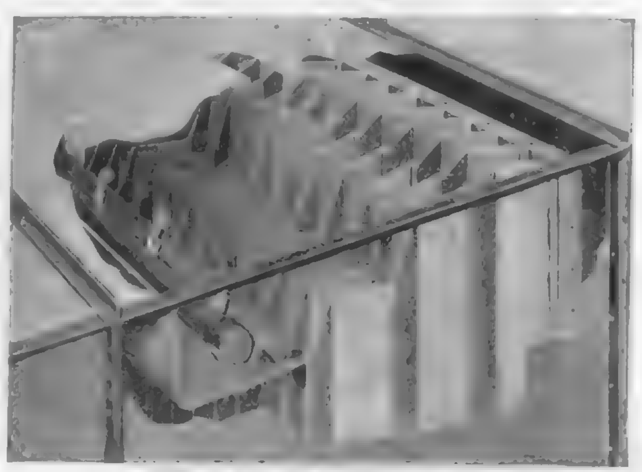
Interior of Carrier Air Washer

The number of nozzles is ample to insure a uniform distribution of the mist as shown in the cut to the left. The water is so finely divided that the air mixes thoroughly with it and all dirt and dust particles are saturated. The air and water then pass through the eliminator plates.

Spray Nozzles in Operation

The eliminators consist of a series of zig-zag plates, a portion of which are flooded with a continuous film of water. The air impinges on these flooded plates, leaving the dust and dirt which are caught in the film of water and washed into the settling tank in the lower part of the washer.

The clean air passes through the dry part of the eliminators where all entrained water is removed by the lips crimped into the plates and leaves the washer with the exact amount of moisture as predetermined by conditions of temperature and humidity.

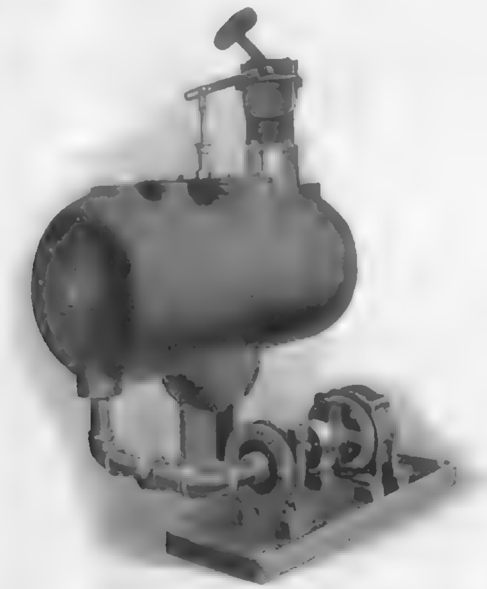


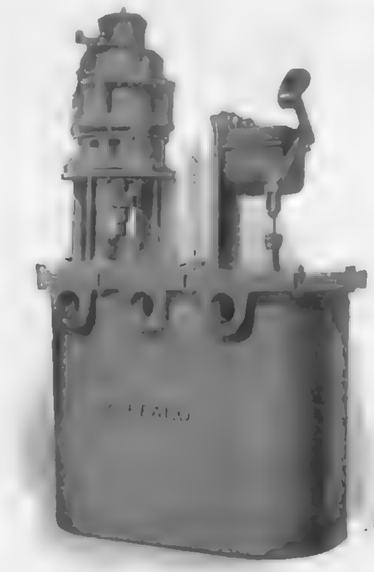
Flooding Nozzles and Eliminators

Turn back to page 17 and see the five pails of dirt removed by the Carrier Air Washer in Public School No. 6, Brooklyn, New York.

Data relative to the sizes and capacities of the Carrier Air Washer and Humidifier will be found on pages 84 to 95.

Fan System of Heating, Ventilating and Humidifying



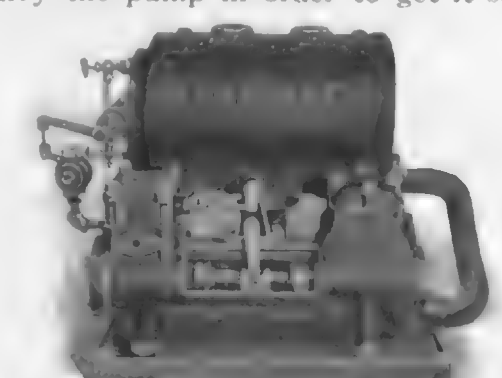


Feed Pumps and Receivers

The Buffalo Feed Pump and Receiver consists of a suitably constructed cast-iron receiving tank, mounted in combination with a Boiler Feed Pump on a common bed plate. The tank is mounted slightly above the pump, giving a sufficient head of water above the suction valves to insure the pump always receiving a full supply of water.

Within the tank is provided a float connected to a chronometer valve controling the steam supply to the pump. Inflowing water causes float to rise, thereby opening the steam supply and starting the pump. When the water level has been lowered, the float automatically cuts off the steam. In this way the condensation water is returned to the boiler as fast as it accumulates.

A Buffalo Vertical Centrifugal Condensation return pump in its scheme of operation is similar in every way to the ordinary horizontal shaft outfit except, that the pump is vertical and submerged within the receiver. The motor is controlled by means of a ten-inch seamless copper float, operating a float switch. This style of design is more convenient in many installations as it avoids providing large pit to carry the pump in order to get it sufficiently low to admit gravity drainage.



The Buffalo Standard Heater

The Buffalo Fan System Pipe Coil Heater has been designed to meet the peculiar requirements found in forced ventilation and also to secure the maximum effectiveness in connection with such work.

First: A perfect circulation of the steam is obtained which removes all air from the coils, carrying it to a single chamber in the base from which it is easily removed through the exhaust connections. Air binding, the greatest enemy of radiation efficiency, is thus prevented.

Second: The heater is so arranged to insure perfect drainage. The design of the base allows no opportunity for pocketing of water, and the pipes are immediately relieved of all condensation, thus avoiding any chance of damage by freezing. The drain ports are made large to allow for an unusually rapid condensation without choking and filling. This feature allows the coil to be used over a great range of radiation.

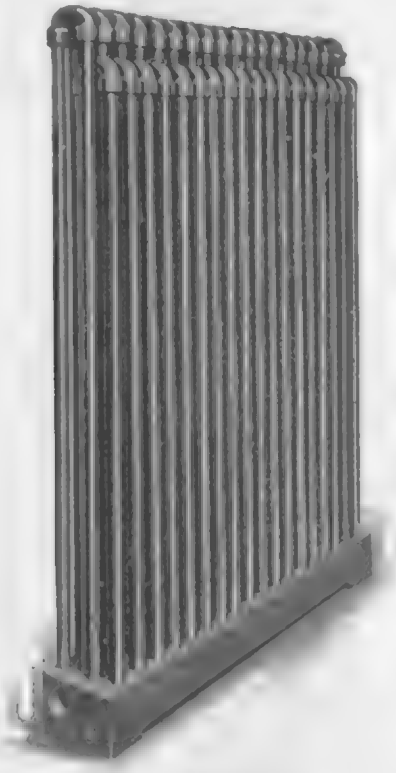
Third: The proportion of the air passages between the coils is so designed as to secure the highest efficiency of radiating surface and the lowest resistance to air flow. In this respect the air is brought in intimate contact with all parts of the heating surface and a uniform and maximum velocity of air is maintained throughout the coil. The velocity of the air is a determining factor in the rate of heat transmission, this being conclusively shown in the curve on page 73, this curve being derived from data obtained from actual tests made on Buffalo Coil Heater. By maintaining uniform velocity through the heater any unnecessary loss of pressure due to changes in velocity is prevented.

Fourth: Each section is independently connected to the steam main and the steam supply controlled by valves so that as few or as many sections as desired may be in operation giving the operator a convenient and absolute control of air temperature and heater effect. By this method of connection any section may be removed for repairs without interfering with the operation of those remaining. This construction also enables the use of live steam in a number of sections and exhaust steam in the remaining, or live and exhaust steam may be introduced in any one section at the same time.

The condensation and heating capacity from a given amount of properly designed radiation, is from three to five times greater with a forced circulation of air than in ordinary radiation. It can readily be seen that a heater designed for a fan system must provide for positive and rapid condensation in order that the coils may be invariably hot. This condition is admirably met with the Buffalo Heater.

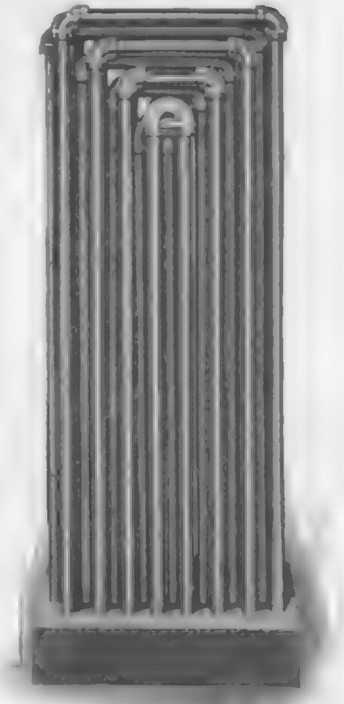
The Buffalo Heater is made in two styles known as the Open Area and Return Bend patterns, the difference being very clearly shown in the cuts on next page.

On pages 96 to 108 are given the tables which show the characteristics of Buffalo Heaters and also various combinations of heaters and fans. This

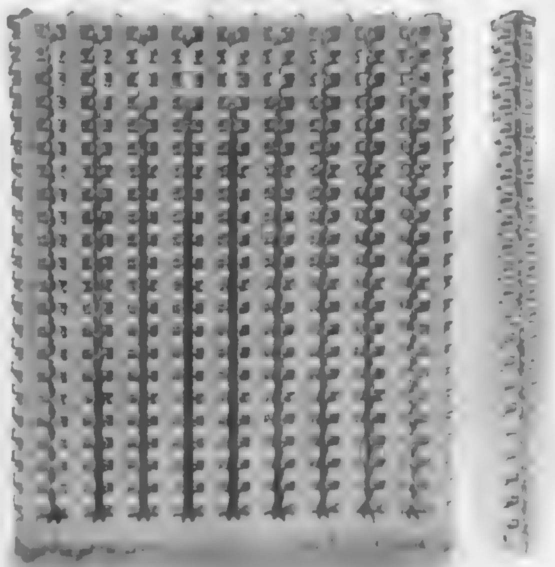


Return Bend Heater

information will be found very useful for use in industrial heating and ventilating work. All Buffalo Heater Sections are made with four rows of pipes. From the table on page 96 it will be seen that the size and number of pipes vary over wide limits so that it is readily possible to obtain a size of heater to meet practically any requirements. When an apparatus is required having a clear area through the coils greater than the largest heater shown in this table, two smaller coils may be chosen and placed back to back, this arrangement can be further extended, and a triplex arrangement of three groups used.



Open Area Heater



Vento Heater

Vento Heaters

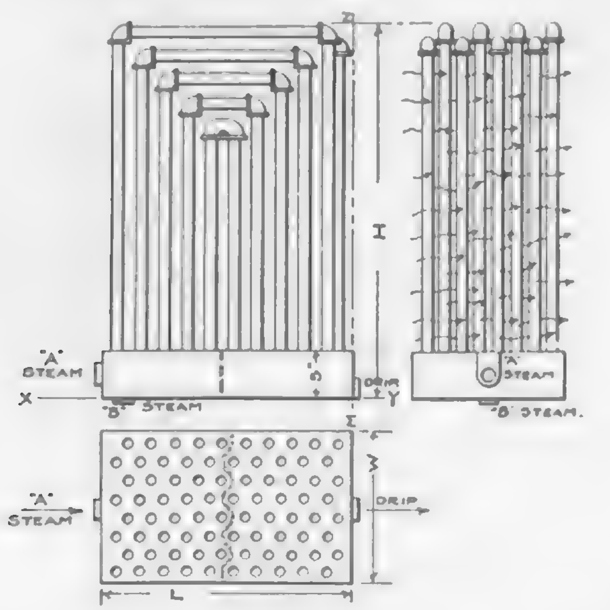
The Vento low pressure cast iron heater, which is very clearly illustrated in the cuts, is designed especially for use in the fan and blower work. These heaters are made in sections of various heights and widths which may be assembled so as to make a heater of any desired size and depth. Ratings are found on page 104.

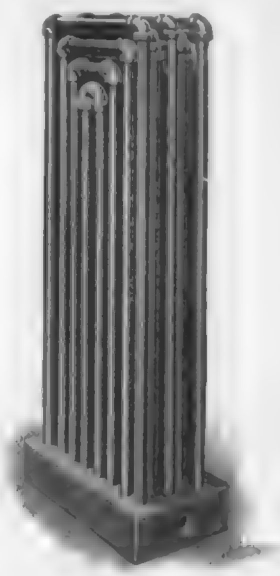
Indirect Heaters

It is sometimes desirable to locate the fan outside of the building to be heated, either in the power house or a specially built apparatus room. If the distance is considerable

it will be found more economical to place the heater unit in the building itself, carrying the unheated air over the intervening space rather than heating it before.

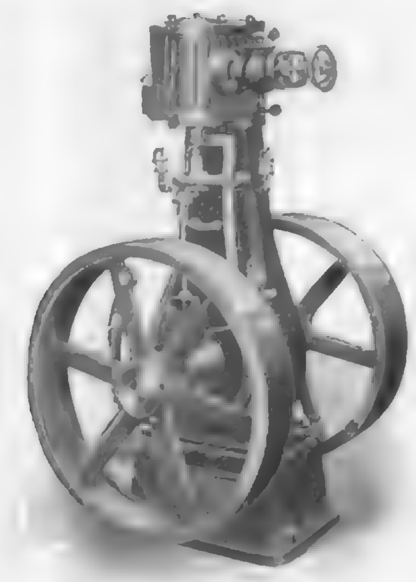
For special indirect heating work where the fan and heater are placed some distance apart a larger base is used for the heater than when used in close proximity to the fan. The table following will give the details of the various sizes of indirect heaters built by this company. Under the heading of "Size" the first row of figures gives the numbers of pipes across the steam supply and drip ends, and the second column the number of pipes in the length of the coil. Cast iron manifold bases are used as in the regular fan system heater, however the steam and exhaust connections are on opposite ends of the manifold instead of on the fan and as in the fan system heater, this enables the heater to be used in either an upright or horizontal position according to the requirements. These heaters are known as the solid base type, the base being divided into two chambers by means of a diaphragm which compels the steam to flow evenly through all pipes. These coils are designed for the use of either live or exhaust steam, being effectively applicable for low pressures.



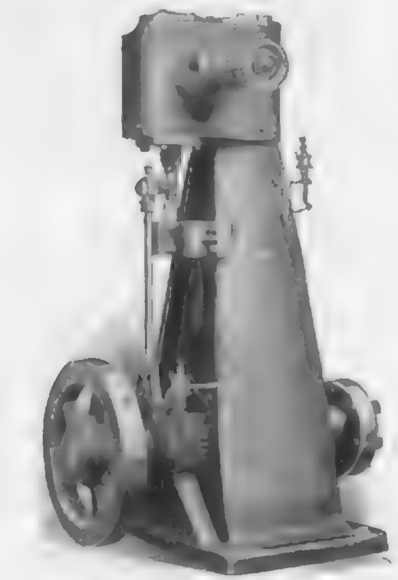


Actual lineal feet one-inch pipe in each section

Size	40½"	46½"	52½"	58½"	64½"	W	L
6 x 8	133	154	177	198	221	12½	22
8 x 8	177	206	236	265	295	16¾	22
8 x 10	221	258	295	332	369	16¾	27
10 x 10	276	323	369	415	462	20	27
10 x 12	346	387	443	498	553	20	32
10 x 14	387	451	517	581	645	20	37
12 x 12	398	464	532	598	663	23¾	32
12 x 14	464	542	618	697	774	23¾	37
12 x 16	532	618	709	798	886	23¾	42
14 x 14	542	632	723	814	906	27½	37
16 x 16	708	827	945	1061	1181	30¼	42



Class "A"



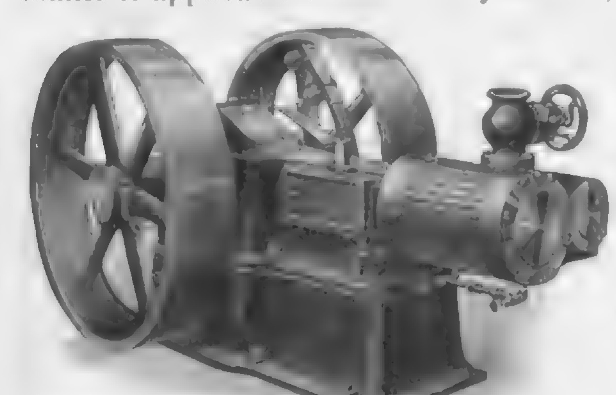
Class "O"

Buffalo Steam Engines

During many years of constant service in the building of engines it has been possible to bring the Buffalo Engine to a high state of perfection. Those who have directed its growth have aimed at the development of a simple, economical and, above all, a substantial engine, built in several types, each suited to its individual work. The limitations of floor spaces and heights, together with different engineering practice in mills and power plants, have been met with appropriate designs which evince a careful consideration of all the requirements.

The design of a steam engine calls for a series of compromises. To make these compromises in favor of the most beneficial results is the evolution of the best engine design, and to carry out these plans in a shop is the evolution of the best engine. Thus it is that the Buffalo Engine has a piston valve and bored guides, that the connecting rod has a small angularity, that the eccentric strap and simple transmission of its motion are used.

The very great extent of the use of the high-speed automatic steam engine makes it applicable to almost any service; and appreciating the fact that there is a demand for these engines of very compact design, giving great power in small space, the construction of the Buffalo Engine, which has been on the market for years, has been constantly improved, and now represents a perfected engine. They are designed to operate with the highest degree of economy. These engines will furnish under the most exacting conditions satisfactory and reliable power.



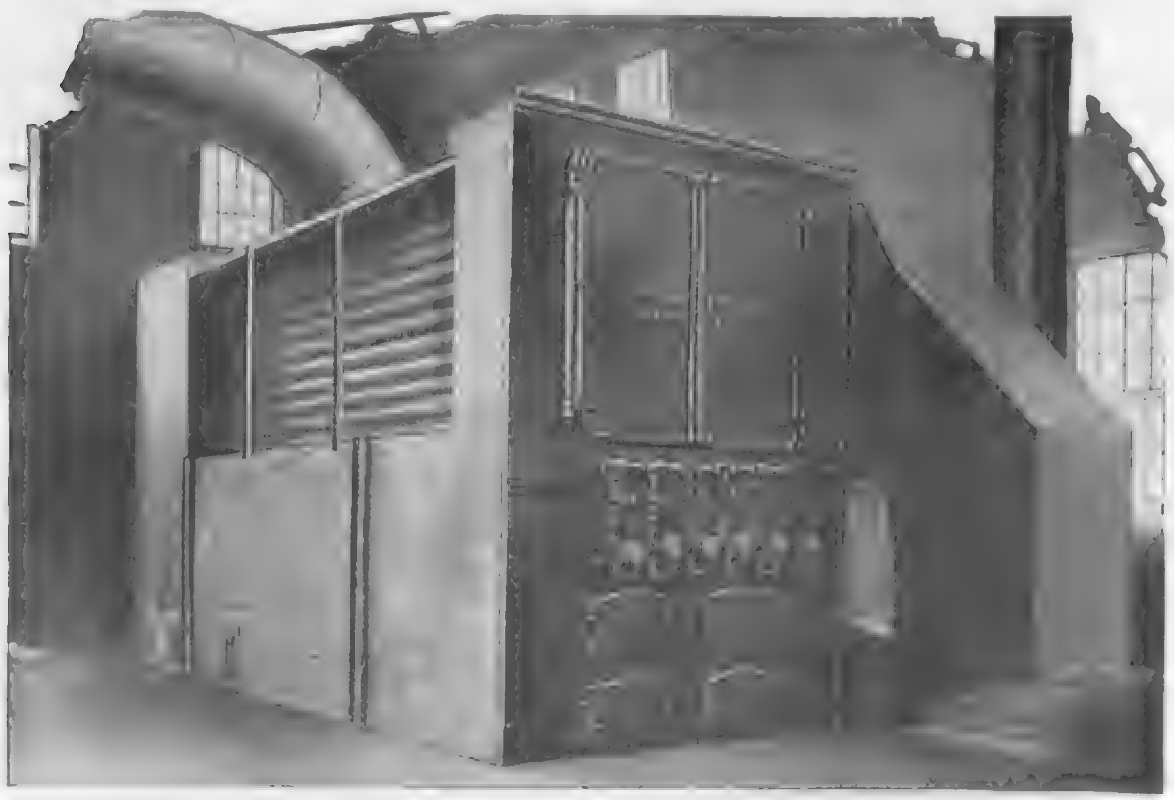
Class "A" Horizontal

Tables of horsepowers and dimensions are given on page 105.

Gas and Coal Heaters

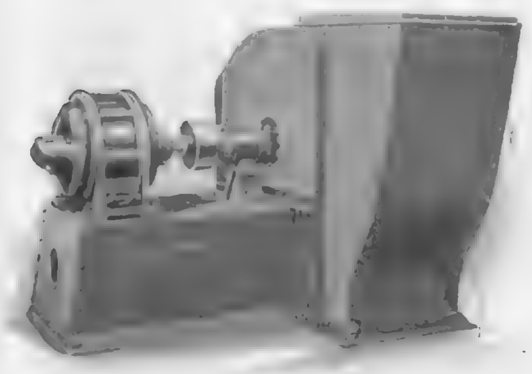
This company has been very successful in the installation of several large heating plants where the heat generated by the combustion of coal or natural gas is transferred direct to the air used for heating and ventilating without the use of an intermediate medium such as water. The heater used in this connection resembles a horizontal water tube boiler, each heater consisting of a bank of iron boiler tubes expanded into a heavy tube sheet at each end. These tubes are set in a brick housing similar to a boiler and the products of combustion passed through the tubes while the air to be heated is passed around the tubes. The furnace and combustion chamber in the housing is so designed that complete combustion will occur before the gases reach the tubes, and thus the greatest possible amount of heat is available for transmission to the air to be heated.

Heaters of this type are in successful operation at the American Rolling Mills, Middletown, Ohio. In these heaters the hot gases at the back of the combustion chamber at a temperature of 3000° F. are mixed with two-thirds of the exhaust gases taken from the front breeching and this resulting mixture is forced through the tubes by means of a fan. This has been found to be the most economical procedure. The gases coming directly from the combustion chamber are too hot to be introduced into the tubes without some cooling and in the method above described no loss of heat is entailed. The pure air for distribution through the building is drawn through the clear area around the outside of the tubes and then forced

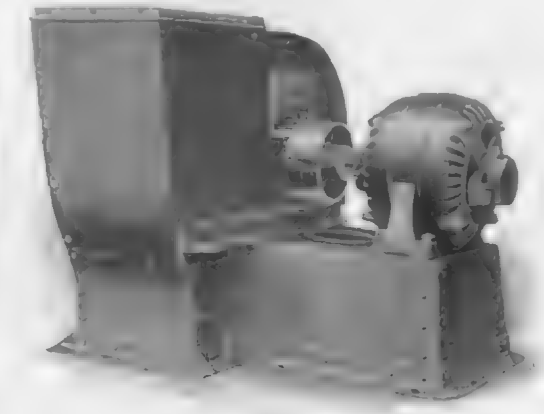


Buffalo Gas Heater at American Rolling Mills, Middletown, Ohio

through the duct system by means of another fan. The heaters mentioned above have been tested and show an average operating efficiency of 85% without considering radiation losses. In many places the heating efficiency obtained by this method makes it advisable to use gas or coal heaters instead of steam boilers. Stokers have been used with great success in heaters of this type.



Overhung Fan



Fan Bearing on Inlet Side

Motor Driven Fans

We have found it advisable in most cases to install engine driven fans, preferably direct connected, this method being most economical and permits of a wide speed variation. There are however, innumerable cases where the steam pressure necessary to operate the engine is not available, or the location desired for the fan apparatus is such that as little attention as possible shall be required for its operation; in cases such as these motor drive affords the solution and special fan designs have been made for use in connection with motors. A motor base is constructed in connection with the fan housing, either of a heavy cast iron one-piece box construction or built up of heavy sheet iron and reinforced with angles. The base is stiffened across the interior by ribs if made of cast iron, or heavy angle braces in the built up construction and made with rounded corners thus combining the necessary strength with a pleasing appearance. In the case of the smaller size of fans with one inlet the fan wheel may be overhung on the motor shaft, which is extended for this purpose; however, it is preferable to use a coupling and place a bearing on the side of the fan farthest from the motor. Wherever alternating current is used, the high speeds at which the regular motors run, make it impossible to use a direct connected unit for heating and ventilating work, except in very rare cases. For direct current, motors may be obtained for any desired speed, and although a slow speed motor is more expensive than a high speed motor of the same power, the advantage gained is sufficient to warrant the adoption of the slow speed motor except in the largest sizes of ventilating fans which operate to best advantage at slow speeds.

Fan Overhung on Motor Shaft

Schenley High School, Pittsburgh, Pa.

Buffalo Equipped

THE BUFFALO FAN SYSTEM OF Heating, Ventilating and Humidifying

PART FOUR

THE Buffalo Forge Company takes great pride in its hand book "Fan Engineering" which is, without exception, the authority in its field. The following subject-matter and data has been condensed from the text of this hand book and the reader is referred to it for a complete discussion of the various principles involved.

Relation of Velocity to Pressure

The laws governing the flow of air are less understood than any other branch of engineering. The flow of air under high pressure must be investigated thermodynamically and the formulae are therefore complicated.

For low pressures such as are met with in ordinary fan work very little error is introduced by applying the same formulae to the flow of air as to the flow of water.

The basic formula for such calculations is

$$V_s = \sqrt{2gh} \quad \text{or} \quad V = 60\sqrt{2gh}$$

where

V_s = velocity in ft. per second.
V = velocity in ft. per minute.
g = acceleration due to gravity in feet per second.
h = Head in feet causing the flow.

We also have

$$U = h' \cdot \frac{d}{12W}$$

where

h' = head expressed in inches of water.
d = density of water.
W = weight of air in pounds per cubic foot.

Then with dry air at 70° F and 29.92″ Barometer, weighing 0.07495 lbs. per cu. foot.

$$\frac{d}{12W} = \frac{62.31}{12 \times 0.07495} = 69.28$$

and we have

$$V = 60\sqrt{2gh' \frac{d}{12W}} = 4005\sqrt{h'}$$

From this we see that the velocity due to one inch of water at standard conditions for air will be 4005 feet per minute and for a pressure of one ounce per square inch will be

$$4005\sqrt{1.734} = 5273 \text{ ft. per minute}$$

The following tables give the pressure and velocity for air first, at constant temperature of 70° and second, at various temperatures.

Corresponding pressures and velocities of dry air at 70° and 29.92 inches barometer

Inches of Water	Ounces per Sq. In.	Velocity Ft. per Min.	Inches of Water	Ounces per Sq. In.	Velocity Ft. per Min.
.05	.0289	896	4.77	2.750	8715
.10	.0577	1266	5.00	2.884	8913
.20	.1154	1791	5.20	3.000	9134
.25	.1443	2003	5.50	3.172	9392
.30	.1730	2193	6.00	3.460	9810
.40	.2308	2533	6.07	3.500	9861
.43	.2500	2637	6.50	3.749	10210
.50	.2884	2832	6.94	4.000	10545
.60	.3460	3102	7.00	4.037	10595
.70	.4037	3351	7.50	4.326	10968
.75	.4326	3468	7.80	4.500	11187
.80	.4614	3582	8.00	4.614	11328
.87	.5000	3729	8.67	5.000	11792
.90	.5190	3800	9.00	5.190	12015
1.00	.5768	4005	9.54	5.500	12367
1.25	.7209	4478	10.00	5.768	12665
1.30	.7500	4566	10.40	6.000	12915
1.50	.8650	4905	11.00	6.344	13282
1.73	1.0000	5273	11.27	6.500	13445
1.75	1.0092	5298	12.00	6.921	13875
2.00	1.1535	5664	12.14	7.000	13950
2.17	1.2500	5895	13.00	7.497	14440
2.25	1.2975	6007	13.87	8.000	14913
2.50	1.4418	6332	14.00	8.074	14985
2.60	1.5000	6457	15.00	8.650	15510
2.75	1.5860	6641	15.61	9.000	15820
3.00	1.7300	6937	16.00	9.227	16020
3.03	1.7500	6976	17.00	9.805	16513
3.25	1.8740	7220	17.34	10.000	16675
3.47	2.0000	7457	18.00	10.380	16990
3.50	2.0185	7492	19.00	10.960	17456
3.75	2.1630	7756	19.07	11.000	17488
3.90	2.2500	7910	20.00	11.535	17910
4.00	2.3070	8010	20.81	12.000	18265
4.25	2.4510	8256	22.54	13.000	19012
4.34	2.5000	8337	24.28	14.000	19730
4.50	2.5950	8496	26.01	15.000	20420
4.75	2.7395	8729	27.74	16.000	21090

Corresponding velocity for dry air at various pressures and temperatures and 29.92 inches barometer

Pressure Inches	Pressure Ounces	50°	60°	70°	100°	150°	300°	500°	550°
.25	.1443	1965	1986	2003	2059	2149	2399	2696	2895
.5	.2884	2778	2808	2832	2911	3038	3391	3812	4095
.75	.4326	3402	3439	3468	3565	3720	4153	4668	5020
1.0	.5768	3929	3971	4005	4117	4296	4796	5390	5795
1.25	.7209	4393	4440	4478	4602	4804	5362	6027	6470
1.50	.8650	4812	4864	4905	5042	5262	5874	6602	7100
1.75	1.0092	5197	5254	5298	5446	5683	6344	7131	7655
2.00	1.1535	5556	5616	5664	5822	6076	6783	7624	8195
2.25	1.2975	5892	5956	6007	6174	6443	7193	8085	8690

"Buffalo"

Some writers have endeavored to correct for the effect of compression by introducing certain constants in the above formulae but the results obtained by the use of these formulae are more in error than when the equations given above are used.

To obtain a more correct formula which will apply to higher pressures up to one-half of an atmosphere, we may assume the air is discharged under isothermal expansion, when we obtain the formula

$$\text{(a)}\quad V_0 = k\sqrt{\frac{1}{d}}\sqrt{\log_{10}\frac{P_0+P}{P_0}}$$

where

P_0 = the barometric pressure in pounds per sq. in.
P = the pressure of the air above atmospheric pressure expressed in inches of mercury.
d = the density in pounds per cu. ft.

If a more exact expression is required, which allows for the adiabatic expansion, the thermodynamic equation is used which gives

$$V_0 = 109.2\sqrt{T_1\left\{1-\left(\frac{P_0}{P_0+P}\right)^{0.29}\right\}}$$

This latter formula is inconvenient in application, and varies so little from formula (a) with pressures under one pound per square inch that formula (a) is always preferable.

Measurement of Air Flow

The quantity, velocity and pressure of air discharged by a fan or flowing through a pipe may be determined by various methods.

The anemometer is used where extreme accuracy is not required or where the velocity of the air is low as in the duct or register entering a room.

Friction of Piping

A subject of great practical importance in fan work is the loss of pressure by friction in conveying air through piping. The expression for the flow of air in smooth circular metal pipes may be taken as approximately

$$F = \frac{1}{50d}\left(\frac{V}{4005}\right)^2$$

where

F = the loss of pressure in inches of water.
V = the velocity in feet per minute.
l = the length of the pipe in feet.
d = the diameter of the pipe in feet, i.e. $\frac{l}{d}$ = length of the pipe in diameters.

From this formula it will be seen that 50 diameters of smooth pipe produce a loss which corresponds to the velocity head. This formula is of the same general

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form developed by Weisbach but recent experiments have shown his coefficient to be considerably too high for smooth pipe and in this formula it has been corrected accordingly. For pipes with rough or uneven surfaces the coefficients must be decreased accordingly. For tile and brick ducts we recommend that the coefficient be decreased 25%.

The tables of pipe friction below will be found very useful in estimating friction losses.

Velocity of air in feet per minute	Loss of Pressure per 100 Ft. in Inches of Water — Diameter of Pipe in Inches													
	3 in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	12 in.	14 in.	16 in.	18 in.	20 in.	22 in.
200	.026	.019	.016	.012	.010	.009	.008	.007	.007	.005	.005	.003	.003	.003
300	.057	.043	.035	.029	.024	.023	.019	.017	.014	.012	.010	.010	.009	.009
400	.102	.076	.062	.050	.043	.038	.033	.031	.026	.022	.019	.017	.016	.014
500	.161	.120	.097	.080	.069	.061	.054	.049	.040	.035	.029	.027	.024	.022
600	.231	.173	.139	.116	.099	.087	.076	.069	.057	.050	.043	.038	.035	.031
700	.314	.239	.189	.158	.135	.118	.104	.094	.078	.068	.059	.052	.047	.043
800	.411	.309	.246	.206	.177	.154	.137	.123	.102	.088	.076	.069	.062	.056
900	.520	.390	.312	.260	.224	.194	.173	.156	.130	.111	.097	.087	.078	.071
1000	.642	.482	.385	.321	.276	.241	.213	.192	.160	.137	.120	.108	.097	.088
1500	1.444	1.083	.867	.723	.619	.541	.482	.434	.361	.312	.277	.243	.225	.198
2000	2.568	1.927	1.542	1.285	1.101	.964	.855	.770	.642	.550	.482	.428	.385	.350
2500	4.013	3.004	2.409	2.006	1.748	1.505	1.337	1.205	1.004	.860	.753	.669	.603	.548
3000	5.774	4.335	3.468	2.890	2.478	2.168	1.927	1.734	1.444	1.238	1.084	.964	.867	.789
3500	7.872	5.902	4.722	3.820	3.373	2.956	2.624	2.360	1.966	1.685	1.476	1.311	1.179	1.073
4000	10.276	7.706	6.166	5.138	4.405	3.853	3.425	3.083	2.568	2.202	1.926	1.713	1.542	1.401
4500	13.005	9.754	7.803	6.560	5.573	4.878	4.335	3.728	3.251	2.787	2.438	2.168	1.951	1.774
5000	16.055	12.051	9.634	8.084	6.880	5.934	5.351	4.852	4.014	3.440	3.010	2.676	2.409	2.190
5500	20.643	14.577	11.656	9.713	8.340	7.288	6.477	5.827	4.857	4.162	3.642	3.237	2.913	2.648
6000	23.120	17.340	13.871	11.561	9.908	8.670	7.706	6.936	5.780	4.985	4.335	3.853	3.468	3.152

Velocity of air in feet per minute	Loss of Pressure per 100 Ft. in Inches of Water — Diameter of Pipe in Inches											
	24 in.	26 in.	28 in.	30 in.	34 in.	38 in.	42 in.	46 in.	50 in.	54 in.	58 in.	62 in.
200	.00322	.00296	.00274	.00257	.00225	.00205	.00184	.00166	.00156	.00139	.00139	.00121
300	.00711	.00668	.00619	.00577	.00510	.00456	.00413	.00376	.00347	.00329	.00295	.00277
400	.01281	.01183	.01099	.01025	.00905	.00810	.00732	.00668	.00607	.00572	.00538	.00486
500	.02005	.01850	.01719	.01604	.01415	.01266	.01146	.01046	.00954	.00884	.00815	.00763
600	.02890	.02667	.02476	.02311	.02039	.01826	.01651	.01491	.01387	.01283	.01179	.01127
700	.03929	.03628	.03388	.03144	.02773	.02481	.02245	.02046	.01873	.01751	.01630	.01526
800	.05134	.04741	.04401	.04108	.03624	.03243	.02934	.02670	.02462	.02289	.02133	.01994
900	.06503	.06003	.05571	.05202	.04590	.04106	.03716	.03399	.03121	.02878	.02688	.02514
1000	.08021	.07404	.06876	.06417	.05661	.05067	.04583	.04214	.03850	.03555	.03312	.03104
1500	.18064	.16677	.15482	.14450	.12750	.11409	.10320	.09427	.08653	.08010	.07473	.06988
2000	.32105	.29638	.27271	.25451	.22460	.20092	.18182	.16732	.15417	.14270	.13282	.12415
2500	.50129	.46300	.42995	.40129	.35402	.31678	.28660	.26167	.24069	.22281	.20740	.19403
3000	.72250	.66695	.61930	.57800	.51000	.45631	.41270	.37680	.34681	.32096	.29895	.27970
3500	.98330	.90761	.84282	.78661	.69415	.62102	.56190	.51295	.47181	.43700	.40680	.38051
4000	1.2841	1.1853	1.1006	1.0274	.90650	.81111	.73381	.66985	.61575	.57096	.53131	.49696
4500	1.6257	1.5051	1.3934	1.3050	1.1476	1.0267	.92899	.84809	.78032	.72135	.67106	.62925
5000	2.0068	1.8525	1.7201	1.5986	1.4166	1.2309	1.1467	1.0462	.96637	.89178	.83022	.77666
5500	2.4284	2.2411	2.0814	1.9426	1.7140	1.5348	1.3873	1.2667	1.1654	1.0791	1.0046	.93980
6000	2.8900	2.6611	2.4771	2.3121	2.0902	1.8252	1.6473	1.5078	1.3872	1.2844	1.1947	1.1167

Sizes of Main and Branch Pipes

Most published rules involve arbitrary constants and tables without giving the basic formula or reasons in determining flue, register and pipe sizes. The most efficient arrangements can be made only when the hypothesis of calculation is understood. The essential data is here given and while its application requires more than merely taking sizes from tables, the whys and wherefores are known, and in this knowledge there is considerable satisfaction.

The piping systems for industrial buildings and those for public buildings are figured according to two distinct methods. In industrial buildings the problem is chiefly to convey the heat units with as great an economy of power, material and space as possible, while in public buildings there are the additional requirements of freedom from noise and prevention of drafts. In industrial buildings the air is usually conveyed through one or two main lines extending lengthwise of the building, the areas of such pipes decreasing as they extend, to give a uniform distribution of air throughout. On the other hand in public buildings, individual ducts are carried from the apparatus to each room, so that it is evident the same method is not applicable to both systems.

Proportioning Pipes in Industrial Buildings

In proportioning the main and branch pipes in industrial buildings, the primary aim is to secure as uniform a distribution as possible without the necessity of damping; secondly, to secure economy of power and economy of material. It has been found good practice in proportioning piping systems to decrease the velocity in the main pipes as the air quantity decreases. This principle of proportioning has three advantages.

First: It utilizes the velocity of the air in producing static pressure in the system.

Second: By this means a nearly uniform static pressure may be secured in all parts of the pipe line, giving a very uniform distribution of air throughout.

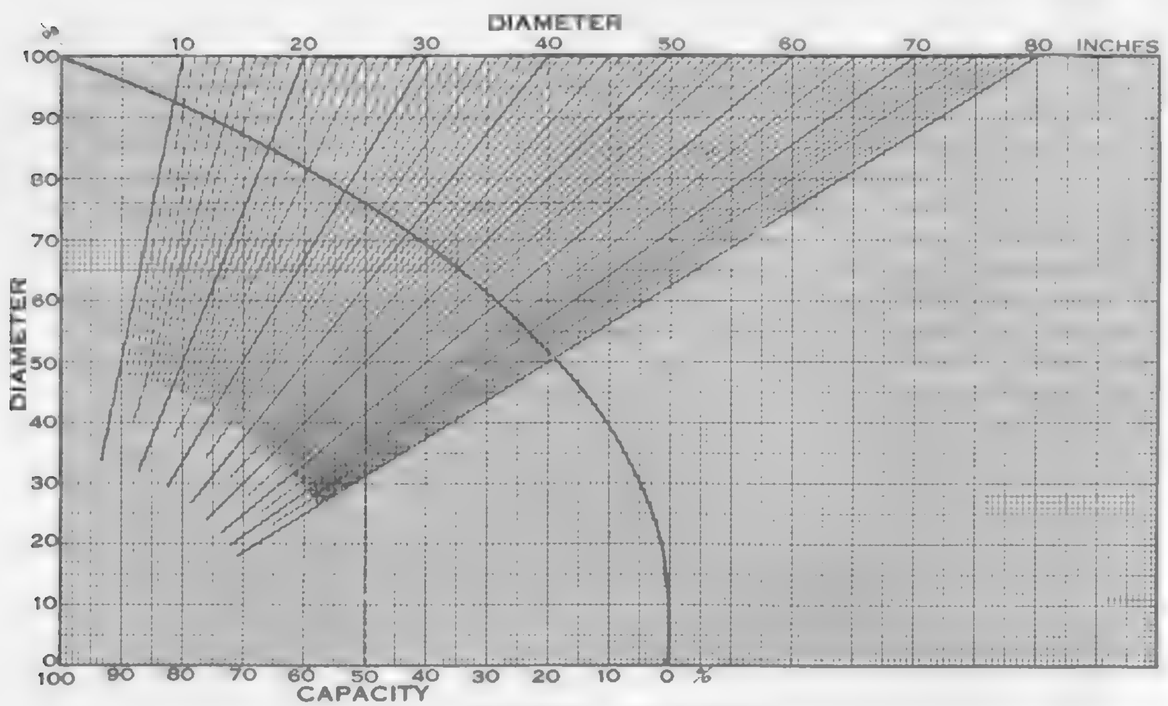
Third: It reduces the friction in the smaller pipes, which would otherwise be excessive.

In carrying out this idea in the proportioning of the piping this company employs an original and accurate method. This method has been carefully tested and has been found to give an almost ideal distribution; the principle involved is to so proportion the velocities in the various pipe sizes as to give equal friction in all air pipes per running foot regardless of their size. It may easily be shown that the equation which relates the carrying capacity of pipe to its size to suit this condition is

$$\frac{d_2}{d_1} = \left(\frac{C_2}{C_1}\right)^{2/5}$$

Where d_1 and d_2 are the relative diameters of two pipes and C_1 and C_2 are the relative carrying capacities. As an equation in this form would be difficult of computation, the chart on page 64 is conveniently employed. In using this chart we start with the main pipe equal in area to the fan outlet, or 10 to 20% larger as circumstances may require. All sizes are proportioned directly from this main pipe

size. It will be noted that the curve is plotted for per cent. capacity and for per cent. diameter according to the formula for constant friction per foot of length. For instance if we have a branch pipe which is required to carry 50% of the capacity of the main pipe, we find the point on the curve which corresponds to 50% capacity and which gives a corresponding point of 76% diameter; that is, a pipe to carry 50% of the capacity with the same friction per foot must have 76% of the diameter, which may be easily calculated or be read directly from the tables for various pipe sizes on page 113. It will be seen that straight lines are drawn for pipe sizes from 20″ up to 80″ in diameter. Supposing the size of the main pipe is 60″ in diameter, then following from left to right along the line of 76% diameter to the line of 60″ pipe we find from the scale above a diameter of 46″, which is the size of pipe which has half the capacity of 60″ pipe with the same friction per foot. By this method the sizes may be read off rapidly without any intermediate figuring whatever.



Proportioning Piping to Allow for Friction

Application

Take the following example which shows the method: Let the main pipe from the fan be 48″ in diameter and suppose a straight duct having ten equal outlets. The first section of piping is 48″, the second section has a capacity of 90%, the third section 80%, the fourth 70%, and so on; corresponding to 90% we find a diameter of 96% which for a 48″ pipe gives us 46″ for the second section. For the third section we have 80% capacity corresponding to 91% diameter or again following from left to right to the 48″ line, we find a diameter of approximately 44″. For the fourth section we have 70% capacity with a corresponding pipe size of 86½% of the main pipe and a diameter of between 41″ and 42″, determined as before. For the last section we have 10% capacity or 40% diameter which gives a diameter of between 19″ and 20″. The outlets may of course be proportioned independently; the same is true of exceptionally long branches which after having been figured in the ordinary way should be increased by a certain percentage throughout as judgment may determine, to decrease the friction.

Determination of Friction

For perfectly smooth, straight galvanized iron pipe it has been found as stated above that the loss of pressure in a length equivalent to 50 diameters is approximately equal to the pressure corresponding to the velocity, i. e., to the velocity head. This holds true for all gases under usual velocities and also for water. In brick and concrete ducts, however, it is advisable to figure 25% more friction or in other words a loss in pressure corresponding to the velocity head for every 40 diameters, i. e., in a 12″ brick duct 40 feet long or 24″ brick duct 80 feet long, the loss in pressure will correspond to the velocity. For instance, 2000 velocity under those conditions will cause a loss in pressure of one-fourth inch. In addition to the above it is necessary to figure the loss in elbows. The factor for elbows is difficult to determine exactly, but from the best information obtainable it appears that one elbow with usual radius is equivalent to a length of pipe of approximately ten diameters.

Now by the foregoing method of proportioning piping, it becomes unnecessary to figure the resistance of each section of pipe independently as the friction is constant per foot of length. It is simply necessary to know the length of the longest run of piping in feet, the number and sizes of elbows and the diameter and velocity in the largest pipe, as the loss is exactly the same as though the entire amount of air was carried through the largest pipe the entire distance. It is usual to figure the area of the main duct approximately equal to the area of the fan outlet. It should be noted that the velocity at the outlet of a Buffalo fan at the rated capacity is equal to one-half of the peripheral velocity, so that the velocity head in the main pipe will be $(\frac{1}{2})^2=\frac{1}{4}$ the total fan pressure. For convenience we may assume the fan to operate at one inch, that the loss in piping thus proportioned is one-fourth inch for every length equal to 40 diameters of the main pipe. As an example of this method of figuring suppose our main outlet is 48″ in diameter and that there are ten sections proportioned as in the previous example. We will also say that the main section contains one elbow, and that there is also an elbow in the section 39″ in diameter, one elbow in the section 30″ in diameter and another elbow in the section 20″ in diameter. Let the length of the pipe to the farthest outlet be 120 feet. We compute the friction in the following way.

120 feet is equivalent to	30 diameters of 48″ pipe.
One 48″ elbow is equivalent to	10 diameters of 48″ pipe.
One 39″ elbow is equivalent to	10 diameters of 39″ pipe or 8.13 diameters of 48″ pipe.
One 39″ elbow is equivalent to	10 diameters of 30″ pipe or 6.25 diameters of 48″ pipe.
One 20″ elbow is equivalent to	10 diameters of 20″ pipe or 4.17 diameters of 48″ pipe.

Total equivalent length 58.55 diameters of 48″ pipe

The equivalent loss in velocity head will then be

$$\frac{58.55}{40} = 1.46$$

times the velocity head in the 48″ main. Further there is the velocity remaining in the 20″ pipe which gives an additional loss evidently of $^{20}/_{48}$ of one velocity head or .42 times the velocity head in the 48″ main. This gives a total loss in the piping system of

$$1.46 + 0.42 = 1.88$$

times the velocity head in the 48″ main. Assuming that the velocity in the 48″ main is 2000 feet per minute corresponding to a velocity head of one-fourth inch, the loss of pressure in the piping system is then

$$0.25 \times 1.88 = .47 \text{ in.}$$

This amount is to be deducted from the total pressure of the fan instead of from the static pressure when the piping is connected directly with the fan outlet, as by the reduction of velocity in the piping we have utilized practically all the velocity pressure at the fan outlet. In a "blow through" apparatus, however, this loss in pressure must be deducted from the static pressure; allowance must likewise be made for the loss in entrance to the piping which may be estimated at 45% of the velocity head. It will thus be seen that a "blow through" system requires larger piping than the "draw through" system for the same results.

In ordinary "draw through" heating system apparatus it is usually advisable to limit the pressure loss in piping to 50% of the total pressure. In the above example it has been shown that 0.47″ out of the total pressure of 1″ is lost if we make the pipe the same size as the fan outlet, and therefore this is safe. However if pressure loss had been 0.65″ and we wished to reduce to 0.5″ we could use the following formula as a loss in pressure varies approximately as the square of the velocity

$$C_2 = C_1\sqrt{\frac{P_2}{P_1}} = C_1\sqrt{\frac{0.50}{0.65}} = 0.88C_1$$

Thus we get the same capacity with .5″ loss as with .65″ loss it would be necessary to increase the area of the piping throughout nearly 13%, or the diameters of all the pipes approximately 6%. Then instead of a 48″ pipe it would be necessary to use a 51″ pipe, inside of a 46″ pipe a 49″ pipe, etc.

Proportioning Ducts for Public Buildings

In public buildings the sizes of air-conveying ducts from fans or heaters to vertical induction flues, and the sizes of these flues, depend upon the velocities of the air flowing in such ducts and flues. The essential factors in determining these velocities are: the limitations of economical rotative speed of fans from the

standpoint of power, the limitations of air velocities on account of noise or by reason of increasing friction as velocities increase; limitation of velocity of inflowing air through registers into rooms; the desirability of as high a velocity of air as is permissible under the limitations referred to in order to get as quick a conveyance of heat units from the heater to the rooms to be heated as possible and to keep down the size of ducts required; and the necessary initial and intermediate velocities to overcome the resistance existing in each particular system.

The size of vertical flues to the registers in the rooms is determined by the maximum velocities allowable in avoiding drafts and noise in the rooms. Practice has shown that the best velocities for the wall registers should be from 200 to 400 feet per minute over the face of the register depending upon the size and location; and for floor registers should be from 125 to 175 feet. The velocity in the vertical flues leading to the registers should be from 400 to 750. The size of these vertical flues is determined largely by the size of register desirable. In general, the velocity in these risers should be low, in order to obtain as uniform a velocity as possible over the register area.

The velocity in the horizontal ducts leading from the apparatus to the vertical risers is determined chiefly by the resistance of the duct. In practice these velocities will vary anywhere from 700 feet to 1200 feet depending upon the size, length of the duct, number of elbows, etc. A designer with considerable experience may proportion these ducts so as to give very uniform distribution without going into any extended calculation. However, it is desirable to have a correct method as a basis. For the benefit of engineers and architects we give here the method employed by this company in the determination of duct velocities and sizes.

The principal losses in piping systems for public buildings are in the horizontal ducts where the velocity is the highest. The losses in these ducts depend upon the velocity, the size and length of duct and upon the number of elbows. There is also considerable loss in pressure as the air enters the duct. An ideal system should take all these factors into consideration, and so proportion the velocities that the resistance would be practically equal in all ducts regardless of the length.

The system which we employ accomplishes this in a practical manner and at the same time avoids any laborious calculation. For each duct a factor may be obtained by inspection in accordance with the following formula:

$$F = 2\tfrac{1}{2} + \frac{L}{4W} + \frac{N}{5}$$

This factor represents the loss by friction in terms of velocity head. The first term, two and one-half, is approximately the number of times the velocity head lost by entrance to the pipe, entrance to the vertical flue, and loss in riser and register. The second factor represents the loss due to length and size of pipe; L is the length in feet and W is the approximate width in inches. The third term represents that proportion of the pressure lost in elbows, and N is the number of long radius elbows. One square elbow is figured equal to two long radius elbows. In checking over the piping layout the factors for the various ducts are first found as above and from these factors the velocity in the respective ducts are ascertained directly. In determining these velocities it is usual to allow a loss not exceeding one-fourth of the total fan pressure. This in practice usually amounts to about one-fourth of an inch. The velocity corresponding to a pressure of one-fourth of an inch is 2000,

and since the velocities vary as the square root of the pressure, the factor F and the velocity V will give a loss of one-fourth of an inch since

$$V = \frac{2000}{\sqrt{F}}$$

In this manner the velocities are accurately and conveniently proportioned.

The Following Table from an Actual Case Illustrates the Variation in Velocities which occur in a Correctly Proportioned System

No. of Rooms	Contents Cubic Feet	Total B. T. U. Loss	A. P. M. Required for Heating	A. P. M. Required for Vent	A. P. M. Allowed	Min. Air Change	A. P. M. for Each Duct	Factor	Velocity in Duct	Area of Duct Sq. Feet
1	5290	13020	260	352	352	15	352	3	950	3.71
2	25700	50380	1008	2570	2570	10	1285	5	730	1.75
3	6070	36240	725	405	760	8	760	6	670	1.14
4	3530	14015	280	235	280	13	280	3	950	.3
5	1860	7985	159	93	159	12	160	3½	880	.19
6	3400	13255	265	227	265	13	265	5	730	.37
7	6070	30370	726	405	726	9	726	7	630	1.16
8	1860	7960	159	93	159	12	150	4	820	.19
9	55400	167090	3340	4440	4440	12½	2220	7	670	3.6

Heating Requirements of Buildings

Before deciding on the heating capacity required, the engineer must make an estimate of the heat losses from the building under the severest conditions of cold weather. The principal loss is by radiation, and as the result of exhaustive tests we have accurate data on the factors for various building materials and types of construction.

The values given on page 109 cover the various types and constructions most frequently met with in ordinary practice. These factors are subject to modification to allow for exposure to winds, unequal distribution of heat, and any extraordinary condition.

The heat required for ventilation is easily computed when the air supplied per hour is known. Since the specified heat of air at constant pressure is 0.238 and the weight of one cubic foot of air at 70° F. is 0.07495 pounds, one British Thermal Unit of heat will raise the temperature of one cubic foot of air

$$\frac{1}{0.238 \times 0.07495} = 56° \text{ F}$$

Infiltration

Loss of heat through infiltration may properly be classed with ventilation losses. It varies greatly with the construction of the building and ranges from one air change in half an hour in a small and poorly constructed building, to one air change in two to three hours in a large well constructed building. This infiltration is caused in part by winds, but chiefly by the chimney-like effect of the column of air in a building at a higher temperature than that outside. The difference in pressure produced is proportional to the difference in temperature and the

amount of infiltration is proportional to the square root of the difference in temperature, hence the heat losses due to infiltration may be expressd by the equation

$$H = C\,(t_2 - t_1)^{3/2}$$

Heater Performance

In modern methods of determining the size of apparatus, whether for heating or drying, the heat losses are first calculated in the manner just described. In public buildings the amount of air is usually specified and the required temperature of air for heating may be determined from the equation

$$t_2 = \frac{k\,l}{0.238 \times 60 \times wa} + t_r$$

in which t_2 = the temperature of the air leaving the heater.

- l = the B. T. U. per hour hour lost by transmission through walls, glass surfaces, roofs, etc.
- a = the cu. ft. of air required for ventilation.
- t_r = the temperature of the room.
- w = weight of one cu. ft. of air (which taken at a temperature of 70° F. and 29.92″ Bar., is 0.07495 lbs.)
- k = is an assumed factor of safety chosen with reference to the particular conditions.

This formula may also be used in determining the volume of air required when the temperature of the air is specified.

Where "return air" is used, that is, air is recirculated from within the building instead of from without, the formula is modified as follows to give the total heat units required with a view of choosing a standard size of apparatus to meet the conditions.

$$H = 0.238\ w\ n\ C\ (t_r - t_1)\ k\ l$$

in which n = number of air changes per hour due to the infiltration of cold air from without. This is dependent upon the size and construction of the building and must be chosen as a result of experiments and tests upon various types of buildings.

- C = the cu. ft. contents of the room.
- t_1 = the outside temperature.

Heating Surface

The next step is to determine the total amount of heating surface in lineal feet of one-inch pipe.

Having previously determined the amount of air to be handled, we determine the size of heater by the free area required to allow the passage of the desired quantity of air at the velocity chosen, according to the following table.

Maximum Velocity Advisable Through Heater for Different Installations

Depth of Heater in Sections	In Public Buildings	In Industrial Plants
4	1140	1500
5	1020	1350
6	930	1230
7	860	1140
8	810	1070

The proper velocity for the air through the clear area of the heater will vary with the different conditions such as pressure carried and character of the installation. The table of velocities given above is based on the assumption that the pressure loss through the heater should not exceed 50% of the total pressure on the fan.

The velocities here given are intended merely to indicate the practical limit, and except where the ducts are very short it will be found advisable to keep below this. This is especially true in the case of public buildings, where the limit should not exceed 90% of the above.

Having determined the velocity through the heater the size of heater required can be readily chosen from the table of sizes and dimensions of Buffalo Standard heaters given on page 96. The same method can be used in connection with the Vento Cast Iron Heater tables given on page 104.

Friction of Heaters

It is even more essential to take account of the friction of the air passing through the heaters than through the piping. The loss of pressure here is much greater than ordinarily imagined and consequently many designers make the mistake of assuming higher velocities than are possible. The following table is compiled from careful tests on Buffalo Heaters.

Friction of Air Through Buffalo Standard Heaters

LOSS OF AIR PRESSURE IN INCHES OF WATER PER SQUARE INCH—AIR AT 70° F.

Velocity Through Clear Area	Number of Sections							
	1	2	3	4	5	6	7	8
300	0.009	0.017	0.026	0.035	0.043	0.052	0.060	0.069
400	0.015	0.034	0.046	0.062	0.077	0.092	0.108	0.123
500	0.024	0.049	0.073	0.095	0.104	0.144	0.168	0.192
600	0.035	0.069	0.104	0.138	0.173	0.207	0.242	0.276
700	0.047	0.094	0.141	0.188	0.235	0.282	0.329	0.376
800	0.061	0.123	0.184	0.245	0.306	0.368	0.429	0.490
900	0.078	0.155	0.233	0.311	0.388	0.466	0.544	0.621
1000	0.096	0.191	0.287	0.382	0.479	0.574	0.670	0.765
1100	0.116	0.232	0.347	0.463	0.579	0.695	0.810	0.926
1200	0.138	0.276	0.414	0.551	0.689	0.827	0.965	1.103
1300	0.162	0.324	0.486	0.648	0.840	0.972	1.133	1.296
1400	0.187	0.375	0.562	0.750	0.936	1.124	1.311	1.500
1500	0.215	0.431	0.646	0.861	1.077	1.293	1.508	1.722
1600	0.245	0.490	0.735	0.980	1.226	1.471	1.716	1.961
1700	0.277	0.555	0.831	1.110	1.387	1.664	1.940	2.218
1800	0.310	0.620	0.930	1.240	1.550	1.860	2.167	2.480

"Buffalo"

The losses are figured for air volumes at 70°. For accurate estimating, correction should be made for the increase in volume due to rise in temperature. The preceding table enables us to read very readily the loss of pressure through the heaters. It is usually advisable to keep the loss in pressure in passing through the heaters down to 50% of the total pressure or less. Therefore for various pressures and various numbers of sections, the figures given in the previous table and based on 50% pressure loss should not be exceeded.

Heater Connection

Care should be taken to have the connection between the fan and the heater case of such a character that it will not restrict the flow of air or offer unnecessary resistance. This precaution is frequently overlooked, either throwing excessive pressure on the fan, or cutting down the quantity of air handled.

The following table gives the approximate lengths of connections advised for draw through installations.

Length of Heater Connection—For Draw Through Equipment

Size Fan Planoidal	Size Fan Nia. and Turbo Conoidal		Distance from Fan to Heater
Up to 70″	Up to No. 7		18″ to 24″
70″ 100″	No. 7	No. 10	24″ to 30″
100″ 130″	No. 10	No. 13	36″
130″ 170″	No. 13	No. 17	42″
170″ 200″	No. 17	No. 20	48″ to 54″

Rate of Condensation

The effect of air velocity and temperature upon the rate of condensation is shown very nicely by the graphical representation of an actual test, on page 72. It will be noted that the rate of transmission decreases with the increase in the temperature of the air in passing through successive sections of the heater but increases very rapidly with the increase in air velocity.

Heater Size

The next step is the determination of the amount of heating surface or the number of heater sections required.

A most convenient method has been devised by our engineers. By means of the curves on page 73 the size of Buffalo Heater can be very readily determined. The use of these curves may best be illustrated by an actual application.

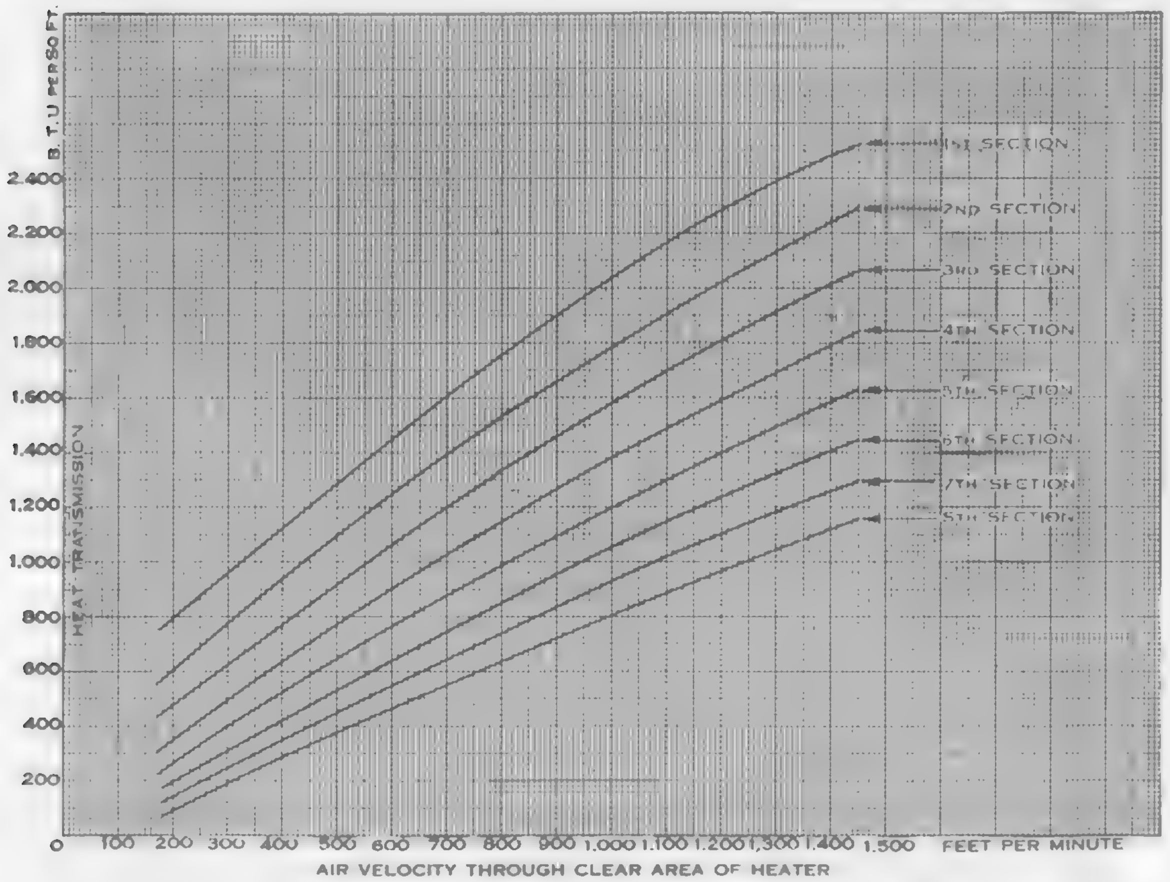
Assume:—Steam pressure on the coils to be 40 pounds, the air to enter at 20° F. and leave at 130° F. and pass through the heater with a velocity of 1000 feet per minute.

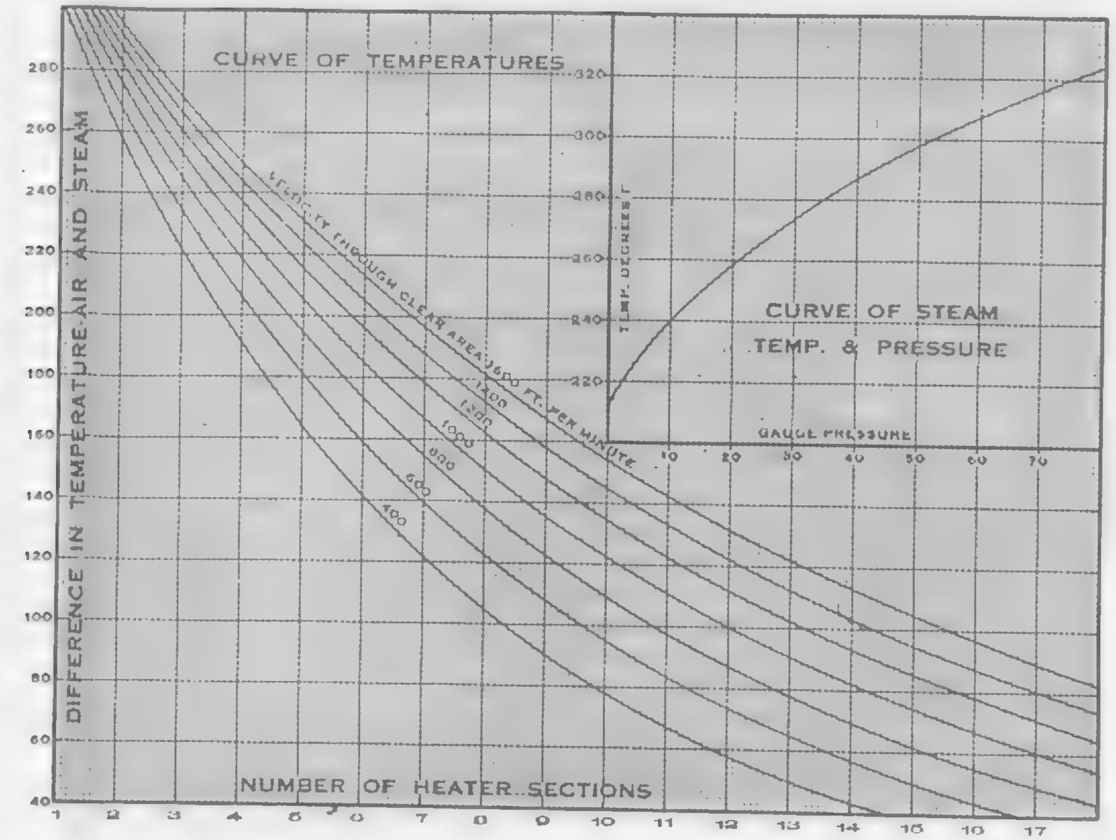
"Buffalo"

From the small curve we see that steam at 40 pound gauge pressure has a temperature of 287° F.

The difference between the temperature of the air entering and the steam will then be 267° and the difference between the air leaving and the steam will be 157°.

Taking the first difference, 267°, and following the line over to the 1000 vel. curve and then down we find 2.55 heater sections. Following the same procedure for the second difference, 157°, we obtain 7.57 sections. The difference between these two results will give the number of sections required which in the case in hand is five.





Condensation in Heater Coils

Having determined the amount of air passing through the heaters and the temperature use of this air the amount of steam condensed per hour can be readily calculated by raising the following formula:

$$C = \frac{a \times (t_2 - t_1) \times 60}{55.2 \times l}$$

When

a = cubic feet or air per minute.
t_1 = temperature of air entering coils.
t_2 = temperature of air leaving coils.
l = latent heat of steam.
55.2 = cubic feet of air raised 1° F. by 1 B.t.u.

Determination of Guarantees

The case often arises that a guarantee to heat a building to a certain specified temperature must be demonstrated at a much higher outside temperature than called for in the guarantee. It then becomes important to know the exact relation between increase in inside temperature when apparatus is operated to its full capacity. This relation has been published for heating with direct radiation, but

it varies considerably from the results obtained with the fan system. Naturally the rise in the indoor temperature will be less than the rise in outdoor temperature owing to the fact that the condensing capacity of the apparatus decreases with the temperature. With a fan system heater the condensing capacity has been shown to be directly proportional to the difference in temperature between steam and air, while with direct radiation it is not directly proportional owing to the variation in convection currents. The same relation between indoor and outdoor temperature may be shown to hold true whether the system was designed to take the air from outdoors entirely or to recirculate air within the building. The formula expressing the relation between indoor and outdoor temperature in either case is,

$$T_r = \frac{T_r'(T_s - T_1) + T_s(T_1 - T_1')}{T_s - T_1'}$$

T_r = Temperature of building obtained with outside temperature T_1.
T_1 = Any outside temperature at which test is made.
T_r' = Temperature of building guaranteed.
T_1' = Specified outside temperature.
T_s = Temperature of steam at pressure specified.

The table following shows corresponding indoor temperatures for various outdoor temperatures with guarantees at 60° to 95° in zero weather.

Table of Average Indoor Temperatures

MAINTAINED AT VARIOUS OUTDOOR TEMPERATURES WITH 5 LBS. STEAM PRESSURE

Outdoor Temp.	Average Indoor Temperatures							
-20	45.2	50.8	56.1	61.6	67.1	72.5	77.9	83.4
-15	48.9	54.3	59.7	64.9	70.3	75.6	80.9	87.3
-10	52.9	57.9	63.1	68.3	73.5	78.7	86.0	89.2
-5	56.3	61.4	66.5	71.6	76.8	81.9	87.0	92.1
0	60°	65°	70°	75°	80°	85°	90°	95°
5	63.7	68.6	73.5	78.4	83.2	88.1	93.0	97.9
10	67.4	72.1	76.9	81.7	86.5	91.3	96.0	100.8
15	71.0	75.7	80.3	85.1	89.7	94.4	99.1	103.7
20	74.7	79.3	83.9	88.4	92.9	97.5	102.1	106.6
25	78.4	82.9	87.3	91.8	96.2	100.7	105.1	109.5
30	82.1	86.4	90.8	94.1	99.4	103.8	108.1	112.4
35	85.8	90.0	94.3	97.5	102.6	106.9	111.2	115.3
40	89.4	93.6	97.7	101.8	105.9	110.0	114.2	118.2
45	93.1	97.1	101.2	105.4	109.1	113.2	117.2	121.1
50	96.8	100.7	104.7	108.5	112.4	116.3	120.2	124.0
55	100.5	104.3	108.1	111.9	115.6	119.4	123.3	126.9
60	104.2	107.8	111.6	115.2	118.8	122.6	126.3	129.8
65	107.8	111.4	115.0	118.6	122.1	125.7	129.3	132.7
70	111.5	115.0	118.5	121.9	125.3	128.8	132.4	135.6

Specimen Problem

To heat a machine shop to 60° F. when 0° outside using all return air from the building. One complete air change every 30 minutes, 20 pound steam pressure at the heaters.

Data

The building consists of three bays 35 feet wide, 245 feet long and 35 feet high, each bay having a saw tooth roof with a pitch of eight feet in the width of the

bay. The floor is of cement with wood above, the walls of brick 17½ inches thick with 20% of the wall area single thickness glass and the roof of paper, tar and gravel laid on two-inch planks.

Solution

Surface	Area	Transmission Factor B.T.U. per 1° per Hour	Transmission Loss B.T.U. per 1° per Hour
Floor	25,720	0.10	2,572
Walls	24,300	0.25	6,075
Glass	6,080	1.00	6,030
Roof	24,460	0.26	6,880

Total cubic contents = 1,003,275 cu. ft.
Infiltration, one air change per hour = 1,003,275 cu. ft.

B.T.U. per 1° = $\frac{1,003,275}{55.2}$ = 18,200

Total B.T.U. per 1° difference = 40,375
Total B.T.U. loss per hour = 40,357 x 60 = 2,421,420
Add 15% margin = 2,784,633

B.T.U. per minute = $\frac{2,784,633}{60}$ = 46,411

Air required per minute = $\frac{1,003,275}{30}$ = 33,442

Final temperature of air leaving heaters = $60 + \frac{46,411 \times 55.2}{33,442}$ = 137° F.

Assume a velocity of 1200 ft. per minute through the clear area of the heater, this will require a heater having

$\frac{33,442}{1200}$ = 27.9 sq. ft. clear area.

From the table on page 96 we find we can use either the 7′-0″x8′-4″ section having 27.2 sq. ft. clear area or the 7′-0″x8′-10″ section having 29.0 sq. ft. clear area.

The first section will give a velocity of

$\frac{33,442}{27.2}$ = 1,230 ft. per minute,

which is close enough to the original assumption of 1200 ft. per minute.

Turning to the table on page 99 we find that with air entering at 60° F. and a velocity of 1200 ft. per minute through the free area of the heater 5 sections of heater will raise the temperature of the air to 143° F. This will decrease slightly due to the actual velocity through the heater being 1230 ft. instead of 1200 ft. per minute.

Let us assume the static resistance of the entire system as two inches and choose a fan to meet our requirments.

From the table on page 78 we find we can use a 120″ planoidal fan which will give 37,050 A.P.M. at 351 R.P.M. by running slightly under rating, or a 110″ planoidal fan which gives 31,000 A.P.M. at 382 R.P.M. by running over rating.

From table on page 79 we can use No. 9 N.C. rated at 35,050 A.P.M. at 364 R.P.M. running under rating.

From table on page 80 we can use No. 9 T.C. rated at 31,800 A.P.M. at 621 R.P.M. running over rating

Buffalo "Baby Conoidal" Fans

Number of Fan	DIMENSIONS: Diameter of Wheel Inches	Diameter of Inlet (Outside) Inches	Diameter of Outlet (Outside) Inches	Revolutions per Minute	Air per Minute Cubic Feet	PRESSURE: Static Inches Water	Total Inches Water	Horse Power	FREE DELIVERY: Air per Minute Cubic Feet	Horse Power
1	4	4	3	1740	80	0.17	0.43	0.012	135	0.030
2	4¾	5½	4	1140	88	0.17	0.25	0.009	150	0.025
				1740	135	0.40	0.60	0.030	230	0.073
3	6⅞	7¾	5¾	1140	260	0.38	0.54	0.050	450	0.120
				1740	400	0.88	1.25	0.180	690	0.440
4	10¼	11¾	8¾	870	700	0.50	0.72	0.11	1200	0.34
				1140	915	0.86	1.23	0.31	1575	0.75
				1440	1155	1.37	1.96	0.63	2000	1.50
				1740	1400	2.00	2.86	1.10	2400	2.65
5	13 1/16	14¼	10⅞	690	1080	0.49	0.71	0.22	1870	0.53
				870	1360	0.78	1.13	0.41	2350	1.00
				1140	1785	1.35	1.94	0.90	3050	2.16
				1440	2255	2.15	3.08	1.81	3880	4.35
6	15⅝	17½	Rectangle 11⅝ x 12⅞	690	1855	0.71	0.98	0.51	3260	1.21
				870	2340	1.13	1.56	1.00	4075	2.36
				1140	3065	1.94	2.67	2.26	5335	5.43
				1440	3875	3.10	4.25	4.55	6740	11.00

Capacities of Buffalo Steel Plate Cone Wheels

Under Average Working Conditions at 70° F. and 29.92" Barometer.

Size	APM per R.P.M. at Free Delivery	¼" Static Pres. R.P.M.	Vol.	H. P.	⅜" Static Pres. R.P.M.	Vol.	H. P.	½" Static Pres. R.P.M.	Vol.	H. P.	¾" Static Pres. R.P.M.	Vol.	H. P.
30	10	393	2,300	0.43	480	2,810	0.79	555	3,250	1.21	680	3,990	2.23
36	17	328	3,330	0.62	400	4,060	1.13	463	4,700	1.75	568	5,760	3.22
42	27	282	4,530	0.85	343	5,530	1.55	396	6,390	2.39	486	7,840	4.39
48	40	246	5,900	1.10	300	7,210	2.02	347	8,350	3.11	425	10,220	5.72
54	57	219	7,480	1.39	266	9,150	2.54	308	10,550	3.92	378	12,950	7.22
60	78	197	9,200	1.71	240	11,250	3.14	278	13,000	4.84	340	15,950	8.90
66	105	178	11,150	2.10	218	13,600	3.83	252	15,750	5.90	309	19,300	10.9
72	136	164	13,300	2.48	200	16,250	4.54	232	18,800	7.00	284	23,050	12.9
84	214	141	18,100	3.38	172	22,100	6.19	199	25,500	9.55	244	31,350	17.6
96	322	123	23,600	4.40	150	28,800	8.07	174	33,350	12.4	213	40,900	22.9
108	459	109	29,950	5.58	133	36,600	10.2	154	42,250	15.8	189	51,900	29.0
120	631	98	36,800	6.85	120	45,000	12.6	138	52,000	19.4	170	63,800	35.6
144	1085	82	53,000	9.90	100	64,850	18.1	116	75,000	28.0	142	91,850	51.5
168	1730	71	72,400	13.5	86	88,450	24.8	100	102,000	38.2	122	125,200	70.2
180	2100	66	83,250	15.5	80	101,800	28.4	93	117,500	43.9	114	144,200	80.6

Fan System of Heating, Ventilating and Humidifying

Buffalo Disc Wheels (Type D)

Size of Fan	Velocity Through Wheel	Cubic Feet of Air per Minute	0.1" S.P. R.P.M.	H. P.	0.2" S.P. R.P.M.	H. P.	0.3" S.P. R.P.M.	H. P.	0.4" S.P. R.P.M.	H. P.	0.5" S.P. R.P.M.	H. P.	0.75" S.P. R.P.M.	H. P.
18"	500	882	739	0.051	871	0.104	978	0.163	1060	0.227	1132	0.30	1297	0.50
	1000	1,762	1100	0.142	1267	0.23	1385	0.32	1477	0.44	1558	0.51	1730	0.77
	1400	2,470	1375	0.281	1535	0.40	1670	0.52	1772	0.64	1870	0.76	2045	1.08
	2000	3,530			1966	0.80	2080	1.00	2200	1.15	2290	1.32	2484	1.68
	2600	4,590					2507	1.73	2600	1.92	2703	2.29	2908	2.67
24"	500	1,570	554	0.091	655	0.185	734	0.29	796	0.41	850	0.53	972	0.88
	1000	3,140	825	0.25	950	0.41	1040	0.57	1108	0.73	1168	0.91	1298	1.38
	1400	4,400	1030	0.50	1150	0.71	1255	0.92	1330	1.14	1402	1.36	1534	1.92
	2000	6,280			1475	1.43	1560	1.77	1650	2.04	1718	2.34	1864	2.99
	2600	8,170					1880	3.08	1950	3.42	2020	4.08	2180	4.76
30"	500	2,450	444	0.142	524	0.29	588	0.45	635	0.63	680	0.83	777	1.37
	1000	4,910	660	0.39	760	0.64	830	0.89	886	1.14	944	1.42	1039	2.15
	1400	6,880	822	0.78	920	1.11	1000	1.43	1062	1.78	1121	2.12	1227	3.00
	2000	9,810			1180	2.24	1247	2.76	1320	3.19	1373	3.65	1491	4.67
	2600	12,767					1505	4.81	1560	5.35	1615	6.38	1745	7.43
36"	500	3,535	369	0.21	436	0.42	488	0.65	530	0.92	566	1.20	648	1.98
	1000	7,060	550	0.57	632	0.92	692	1.28	740	1.64	778	2.04	865	3.40
	1400	9,900	687	1.13	765	1.60	837	2.06	888	2.56	935	3.05	1024	4.32
	2000	14,150			984	3.22	1040	3.98	1100	4.60	1115	5.27	1243	6.73
	2600	18,340					1255	6.92	1300	7.70	1348	9.20	1454	10.71
42"	500	4,808	316	0.28	374	0.57	420	0.89	456	1.24	486	1.63	556	2.69
	1000	9,616	472	0.77	544	1.25	594	1.74	632	2.24	668	2.78	742	4.21
	1400	13,475	588	1.53	656	2.17	718	2.80	760	3.48	802	4.15	876	5.88
	2000	19,232			844	4.37	892	5.43	944	6.25	982	7.17	1066	9.15
	2600	25,025					1074	9.43	1114	10.45	1154	12.50	1246	14.58
48"	500	6,280	277	0.36	327	0.74	367	1.16	398	1.62	425	2.13	486	3.52
	1000	12,560	412	1.01	475	1.63	520	2.27	554	2.92	584	3.63	649	5.50
	1400	17,600	515	2.00	575	2.84	627	3.67	665	4.56	701	5.42	767	9.68
	2000	25,120			737	5.72	780	7.08	825	8.16	859	9.35	932	11.96
	2600	32,680					910	12.32	975	13.68	1010	16.33	1090	19.04
54"	500	7,948	246	0.46	291	0.94	326	1.47	354	2.05	377	2.70	432	4.46
	1000	15,890	366	1.28	422	2.07	461	2.88	491	3.72	518	4.60	577	6.96
	1400	22,275	457	2.54	510	3.60	557	4.66	590	5.77	623	6.86	682	9.72
	2000	31,795			655	7.22	694	8.95	734	10.30	763	11.85	828	15.14
	2600	41,360					856	15.60	866	17.30	898	20.65	968	21.10
60"	500	9,812	221	0.57	262	1.16	293	1.81	319	2.52	340	3.33	389	5.50
	1000	19,625	330	1.58	380	2.55	416	3.54	443	4.57	467	5.67	519	8.60
	1400	27,500	412	3.13	460	4.44	502	5.74	532	7.13	561	8.53	614	12.00
	2000	39,250			590	8.94	624	11.10	660	12.80	688	14.62	716	18.70
	2600	51,081					751	19.30	780	21.4	808	25.5	872	29.8
72"	500	14,130	185	0.82	218	1.66	245	2.61	265	3.64	283	4.80	323	7.92
	1000	28,260	275	2.26	317	3.67	347	5.12	370	6.58	389	8.16	433	12.38
	1400	39,600	343	4.50	383	6.40	417	8.26	442	10.20	468	12.20	512	17.29
	2000	56,520			490	12.90	520	15.90	550	18.30	573	21.1	622	26.9
	2600	73,530					627	27.7	650	30.8	671	36.8	727	42.8
84"	500	19,232	158	1.11	187	2.27	210	3.55	228	4.95	243	6.54	278	10.78
	1000	38,464	236	3.09	272	5.00	297	6.95	316	8.95	334	11.11	371	16.85
	1400	53,900	294	6.13	328	8.68	359	11.20	380	13.90	401	16.61	438	23.5
	2000	76,930			422	17.50	446	21.70	472	25.00	491	28.7	533	36.6
	2600	100,100					537	37.7	557	41.8	577	50.0	624	58.4

Capacities of Buffalo Planoidal Steel Plate Blowers (Type L) Under Average Working Conditions

70° F. 29.92" Barometer

Size	Diameter of Blast Wheel Inches	Area of Outlet Square Ft.	½" Static Pressure = 0.288 Ounces			¾" Static Pressure = 0.433 Ounces			1" Static Pressure = 0.577 Ounces			1½" Static Pressure = 0.865 Ounces		
			R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.
30	19¼	0.77	678	1,360	0.23	850	1,420	0.42	958	1,640	0.65	1174	2,010	1.19
35	22½	1.04	580	1,570	0.31	710	1,925	0.58	820	2,220	0.87	1005	2,720	1.63
40	25¾	1.36	508	2,065	0.41	623	2,530	0.75	719	2,920	1.16	880	3,580	2.13
45	29⅞	1.75	451	2,600	0.52	554	3,185	0.96	639	3,680	1.47	783	4,510	2.70
50	32¼	2.16	407	3,220	0.64	498	3,940	1.18	575	4,550	1.82	705	5,580	3.35
55	35⅜	2.61	369	3,890	0.77	452	4,765	1.42	522	5,500	2.19	640	6,740	4.03
60	38½	3.13	339	4,630	0.92	415	5,675	1.70	479	6,550	2.61	587	8,030	4.80
70	45	4.26	290	6,320	1.25	355	7,730	2.31	410	8,930	3.55	502	10,920	6.52
80	51⅛	5.54	254	8,230	1.64	315	10,080	3.02	359	11,630	4.65	440	14,250	8.55
90	57⅞	7.10	226	10,440	2.08	276	12,750	3.82	319	14,730	5.88	391	18,050	10.80
100	64¼	8.75	203	12,880	2.56	248	15,750	4.71	287	18,200	7.25	352	22,300	13.32
110	70¾	10.57	185	15,550	3.10	226	19,100	5.71	261	22,000	8.78	320	26,950	16.12
120	77¼	13.00	169	18,530	3.69	207	22,700	6.78	239	26,200	10.44	293	32,080	19.18
130	83½	14.85	156	21,600	4.31	192	26,150	7.93	221	30,550	12.20	271	37,140	22.40
140	90	17.20	145	25,200	5.02	177	30,850	9.24	205	35,650	14.20	251	43,700	26.10
150	96½	19.70	135	28,950	5.76	165	35,400	10.60	191	40,900	16.30	234	50,150	29.95
160	103	22.40	127	32,800	6.57	154	40,200	12.10	179	46,450	18.60	219	56,900	34.15
170	109¼	25.40	120	37,150	7.42	146	45,500	13.65	169	52,550	21.00	207	64,400	38.60
180	115¾	28.50	112	41,700	8.31	138	51,100	15.25	159	59,000	23.50	195	72,250	43.15
190	122¼	31.70	107	46,300	9.26	131	56,700	17.05	151	65,500	26.20	185	80,250	48.10
200	128½	35.30	102	51,500	10.25	125	63,100	18.85	144	72,850	29.00	176	89,200	53.30

Size	Diameter of Blast Wheel Inches	Area of Outlet Square Ft.	2" Static Pressure = 1.154 Ounces			2½" Static Pressure = 1.442 Ounces			3" Static Pressure = 1.734 Ounces			3½" Static Pressure = 2.019 Ounces		
			R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.
30	19¼	0.77	1355	2,320	1.84	1515	2,595	2.57	1660	2,840	3.38	1792	3,070	4.26
35	22½	1.04	1160	3,140	2.52	1295	3,510	3.48	1420	3,845	4.63	1534	4,155	5.83
40	25¾	1.36	1018	4,135	3.28	1135	4,620	4.58	1245	5,060	6.03	1345	5,460	7.60
45	29⅞	1.75	904	5,210	4.15	1010	5,825	5.81	1108	6,375	7.63	1195	6,890	9.63
50	32¼	2.16	814	6,440	5.15	910	7,200	7.20	996	7,880	9.45	1076	8,510	11.91
55	35⅜	2.61	738	7,780	6.19	826	8,700	8.66	904	9,530	11.38	976	10,290	14.34
60	38½	3.13	678	9,260	7.38	758	10,370	10.31	830	11,340	13.55	896	12,250	17.10
70	45	4.26	580	12,630	10.02	648	14,120	14.03	710	15,460	18.45	767	16,700	23.25
80	51⅛	5.54	508	16,450	13.12	568	18,400	18.40	621	20,150	24.20	672	21,750	30.50
90	57⅞	7.10	451	20,850	16.60	505	23,300	23.30	553	25,500	30.55	597	27,550	38.50
100	64¼	8.75	406	25,750	20.48	454	28,800	28.70	497	31,530	37.70	537	34,050	47.50
110	70¾	10.57	369	31,100	24.80	413	34,800	34.70	452	38,100	45.60	488	41,200	57.50
120	77¼	13.00	338	37,050	29.50	378	41,400	41.30	414	45,400	54.25	447	49,000	68.40
130	83½	14.85	313	43,250	34.50	350	48,350	48.25	383	52,900	63.40	413	57,200	80.00
140	90	17.20	290	50,400	40.15	324	56,100	56.15	355	61,750	73.80	384	66,700	93.00
150	96½	19.70	270	57,900	46.10	302	64,750	64.50	331	70,900	84.70	358	76,600	106.80
160	103	22.40	253	65,700	52.60	284	73,500	73.50	310	80,400	96.60	335	86,900	121.80
170	109¼	25.40	239	74,300	59.40	267	83,200	83.00	293	91,000	109.00	316	98,400	137.50
180	115¾	28.50	225	83,500	66.40	251	93,400	93.00	277	102,200	122.30	298	110,400	154.00
190	122¼	31.70	214	92,650	74.20	239	103,700	103.60	262	113,300	136.40	282	122,700	171.50
200	128½	35.30	204	104,000	82.00	228	115,400	114.70	250	126,400	150.80	269	136,300	190.00

Total Pressure is 126% of the Rated Static Pressure.

Capacities of Buffalo Niagara Conoidal Fans—(Type N) Under Average Working Conditions

70° F and 29.92" Barometer

Size	Diameter of Blast Wheel Inches	Area of Outlet Square Ft.	½" Static Pressure = 0.288 Ounces			¾" Static Pressure = 0.433 Ounces			1" Static Pressure = 0.577 Ounces			1½" Static Pressure = 0.865 Ounces		
			R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.
3	15¾	1.31	544	1,915	0.28	668	2,380	0.51	770	2,750	0.78	943	3,365	1.45
3½	18¼	1.79	465	2,642	0.48	572	3,240	0.69	660	3,740	1.06	809	4,580	1.97
4	20½	2.33	408	3,459	0.50	500	4,290	0.90	577	4,895	1.39	709	5,980	2.58
4½	23½	2.95	362	4,375	0.63	445	5,350	1.11	514	6,195	1.75	630	7,575	3.26
5	26¼	3.64	326	5,400	0.77	400	6,610	1.41	462	7,645	2.16	566	9,350	4.03
5½	28¾	4.41	296	6,540	0.94	364	8,000	1.71	420	9,250	2.62	515	11,320	4.87
6	31½	5.25	272	7,780	1.11	334	9,525	2.03	386	11,000	3.12	472	13,450	5.80
7	36½	7.14	233	10,590	1.52	286	12,950	2.77	330	14,980	4.24	405	18,330	7.90
8	42	9.33	204	13,820	1.98	250	16,910	3.61	289	19,550	5.54	354	23,950	10.30
9	47	11.81	181	17,500	2.51	222	21,400	4.57	256	24,750	7.01	314	30,300	13.05
10	52	14.58	163	21,600	3.09	200	26,450	5.65	231	30,550	8.65	283	37,400	16.10
11	58	17.64	148	26,150	3.74	182	32,000	6.85	210	37,000	10.48	257	45,250	19.18
12	63	21.00	136	31,100	4.45	167	38,100	8.15	193	44,050	12.48	236	53,900	23.20
13	68	24.65	125	36,500	5.22	154	44,700	9.56	178	51,650	14.62	217	63,200	27.20
14	73	28.68	116	42,350	6.06	143	51,900	11.08	165	60,000	16.96	202	73,200	31.55
15	78	32.80	109	48,550	6.95	133	59,500	12.70	154	68,850	19.49	189	84,100	36.25
16	84	37.32	102	55,300	7.91	125	67,750	14.46	144	78,300	22.15	177	95,750	41.20
17	89	42.14	96	62,500	8.95	118	76,500	16.32	136	88,400	25.00	167	108,000	46.50
18	94	47.24	91	70,000	10.01	111	85,600	18.30	128	99,100	28.05	157	121,200	52.15
19	99	52.63	86	78,000	11.15	105	95,500	20.40	122	110,200	31.25	149	135,000	58.05
20	105	58.32	82	86,450	12.36	100	105,850	22.60	116	122,200	34.65	142	149,500	64.45

Size	Diameter of Blast Wheel Inches	Area of Outlet Square Ft.	2" Static Pressure = 1.154 Ounces			2½" Static Pressure = 1.442 Ounces			3" Static Pressure = 1.734 Ounces			3½" Static Pressure = 2.019 Ounces		
			R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.	R.P.M.	Volume Cubic Ft. per Min.	H.P.
3	15¾	1.31	1088	3,890	2.21	1245	4,350	3.08	1332	4,770	4.05	1443	5,150	5.13
3½	18¼	1.79	934	5,300	3.01	1010	5,930	4.19	1141	6,495	5.53	1238	7,010	6.98
4	20½	2.33	817	6,920	3.93	912	7,730	5.47	1000	8,480	7.22	1082	9,160	9.12
4½	23½	2.95	726	8,750	4.97	810	9,795	6.93	890	10,740	9.14	964	11,590	11.55
5	26¼	3.64	655	10,820	6.15	730	12,070	8.55	800	13,250	11.26	868	14,300	14.25
5½	28¾	4.41	595	13,100	7.43	664	14,600	10.35	728	16,030	13.62	789	17,300	17.25
6	31½	5.25	545	15,550	8.85	609	17,390	12.30	667	19,000	16.22	723	20,600	20.55
7	36½	7.14	468	21,200	12.02	522	23,650	16.75	572	26,000	22.10	620	28,050	27.95
8	42	9.33	409	27,650	15.70	456	30,900	21.90	500	33,950	28.85	542	36,600	36.50
9	47	11.81	364	35,050	19.90	405	39,100	27.70	445	42,950	36.55	482	46,350	46.20
10	52	14.58	327	43,250	24.55	365	48,300	34.20	400	53,000	45.15	433	57,200	57.00
11	58	17.64	297	52,300	29.70	332	58,450	41.45	364	64,100	54.60	394	69,300	69.00
12	63	21.00	272	62,300	35.50	304	69,550	49.25	334	76,400	65.00	361	82,500	82.15
13	68	24.65	252	73,050	41.50	280	81,600	57.80	308	89,550	76.30	334	96,750	96.45
14	73	28.68	234	84,900	48.15	261	94,600	67.05	286	103,900	88.70	310	112,050	111.90
15	78	32.80	218	97,250	55.25	243	108,700	77.00	267	119,200	101.50	289	128,800	128.20
16	84	37.32	204	110,750	62.85	228	123,600	87.50	250	135,800	115.60	271	146,100	146.00
17	89	42.14	192	125,000	71.00	214	139,500	99.00	235	153,100	130.30	255	165,300	164.80
18	94	47.24	182	140,000	79.50	203	156,500	110.80	222	171,800	146.00	241	185,300	184.60
19	99	52.63	172	156,000	88.55	192	174,200	123.40	211	191,200	162.80	228	206,200	206.00
20	105	58.32	164	173,000	98.25	183	193,000	136.80	200	212,000	180.30	217	229,000	228.00

Total Pressure is 127.4% of the Rated Static Pressure.

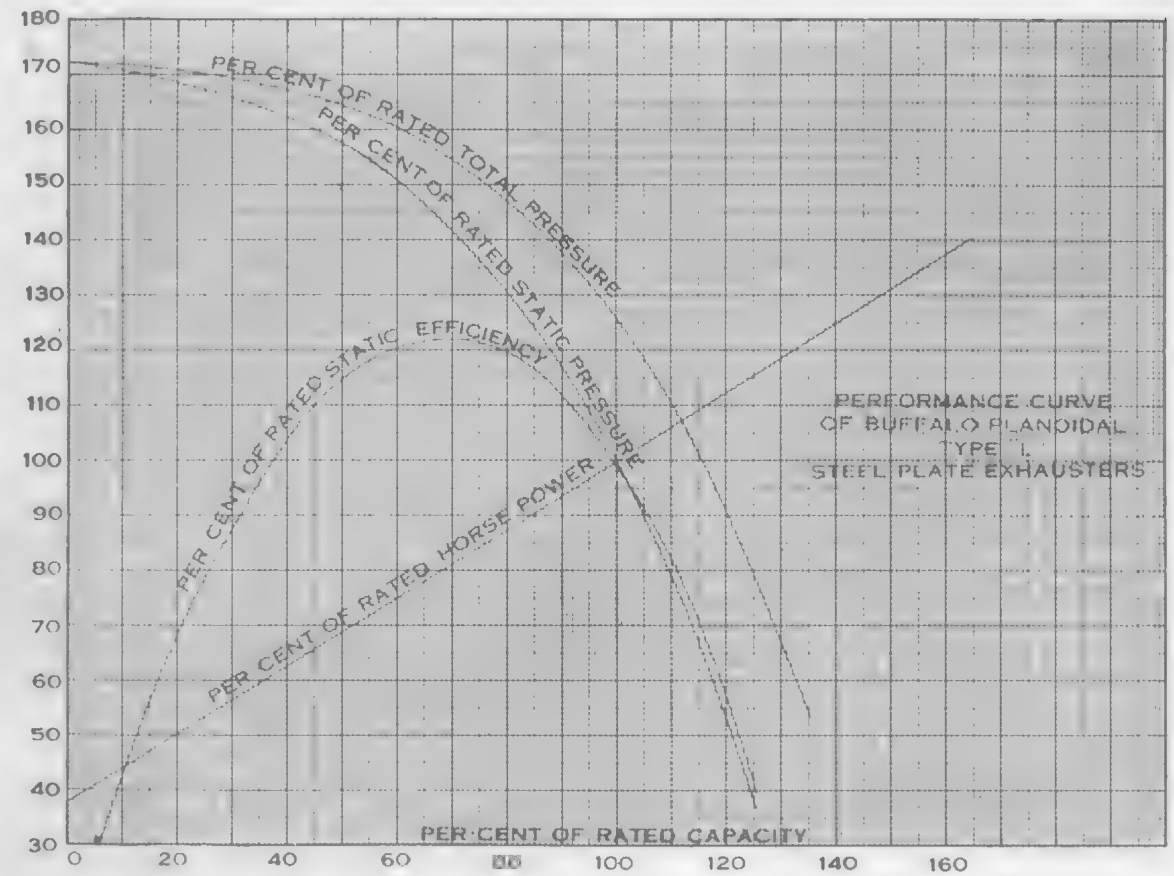
Capacities of Buffalo Turbo Conoidal Fans (Type T) Under Average Working Conditions

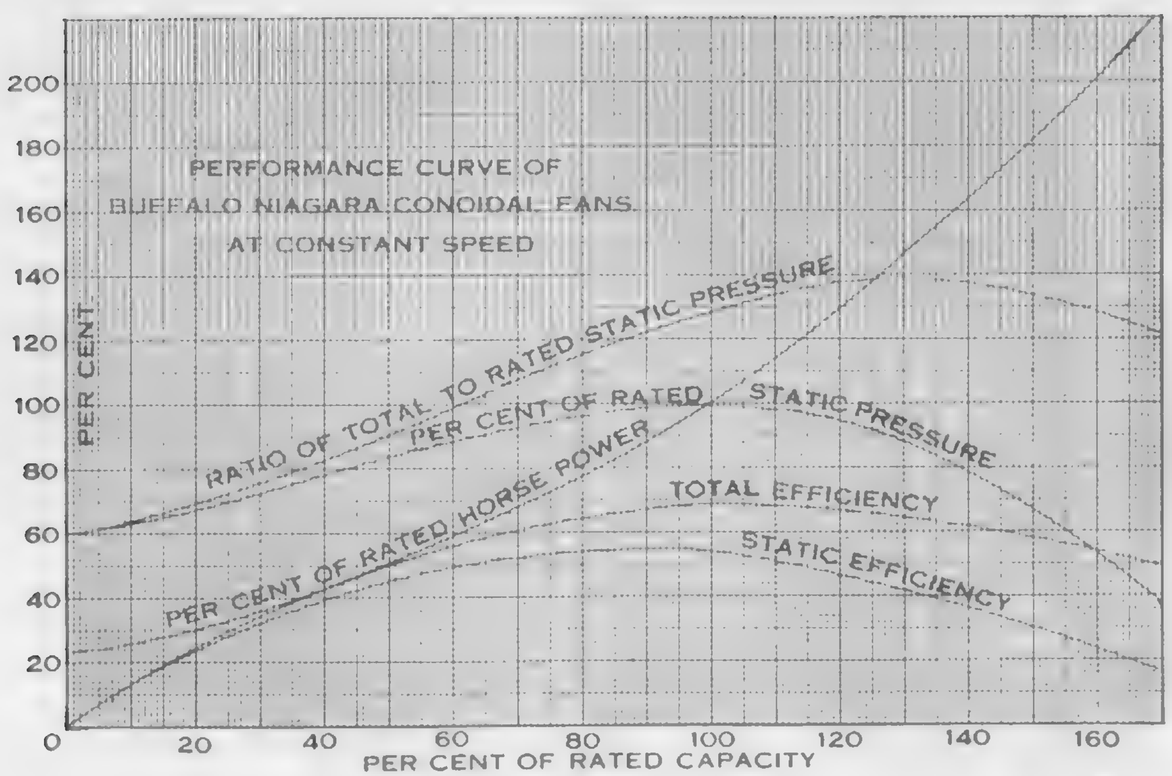
70° F. and 29.92" Barometer

Size	Diameter of Blast Wheel Inches	Area of Outlet Square Ft.	½" Static Pressure = 0.288 Ounces R.P.M.	Volume Cubic Ft. per Min.	H.P.	¾" Static Pressure = 0.433 Ounces R.P.M.	Volume Cubic Ft. per Min.	H.P.	1" Static Pressure = 0.577 Ounces R.P.M.	Volume Cubic Ft. per Min.	H.P.	1½" Static Pressure = 0.865 Ounces R.P.M.	Volume Cubic Ft. per Min.	H.P.
2½	14¼	0.91	1,115	1,230	0.20	1,368	1,500	0.36	1,580	1,740	0.56	1,935	2,120	1.03
3	17¼	1.31	930	1,770	0.28	1,140	2,160	0.52	1,315	2,500	0.81	1,610	3,060	1.48
3½	20	1.79	797	2,410	0.39	976	2,940	0.71	1,130	3,410	1.10	1,380	4,160	2.02
4	22¾	2.33	697	3,140	0.51	855	3,850	0.93	987	4,450	1.44	1,208	5,440	2.64
4½	25¾	2.95	620	3,980	0.64	760	4,860	1.18	879	5,640	1.82	1,075	6,890	3.34
5	28½	3.64	558	4,910	0.79	684	6,000	1.45	790	6,950	2.25	966	8,500	4.12
5½	31½	4.41	507	5,950	0.96	621	7,270	1.76	719	8,400	2.72	880	10,300	5.00
6	34¼	5.25	465	7,070	1.14	570	8,650	2.09	658	10,000	3.24	806	12,250	5.77
6½	36¾	6.16	430	8,300	1.33	526	10,200	2.46	608	11,750	3.80	743	14,370	6.96
7	39¾	7.14	398	9,630	1.55	488	11,780	2.85	565	13,610	4.40	690	16,650	8.09
7½	42½	8.19	372	11,050	1.78	456	13,500	3.27	526	15,610	5.05	645	19,100	9.27
8	45½	9.33	349	12,590	2.02	428	15,370	3.72	495	17,800	5.75	604	21,750	10.55
8½	48	10.53	328	14,200	2.28	402	17,580	4.21	465	20,100	6.50	569	24,600	11.90
9	51¼	11.81	310	15,900	2.56	380	19,450	4.71	440	22,500	7.29	536	27,500	13.35
10	56¾	14.58	279	19,650	3.16	342	24,050	5.82	395	27,800	9.00	483	34,000	16.50
11	62½	17.64	254	23,800	3.82	311	29,100	7.05	359	33,700	10.90	439	41,100	19.95
12	68	21.00	232	28,300	4.55	286	34,600	8.40	329	40,100	12.95	402	49,000	23.80
13	73½	24.65	214	33,200	5.34	263	40,600	9.85	304	47,000	15.20	372	57,500	27.90
14	79	28.68	198	38,500	6.20	244	47,100	11.40	282	54,500	17.62	345	66,700	32.35
15	84¾	32.80	186	44,200	7.11	228	54,050	13.08	264	62,600	20.20	322	76,500	37.15
16	90¼	37.32	174	50,300	8.09	214	61,500	14.90	247	71,200	23.00	302	87,100	42.25

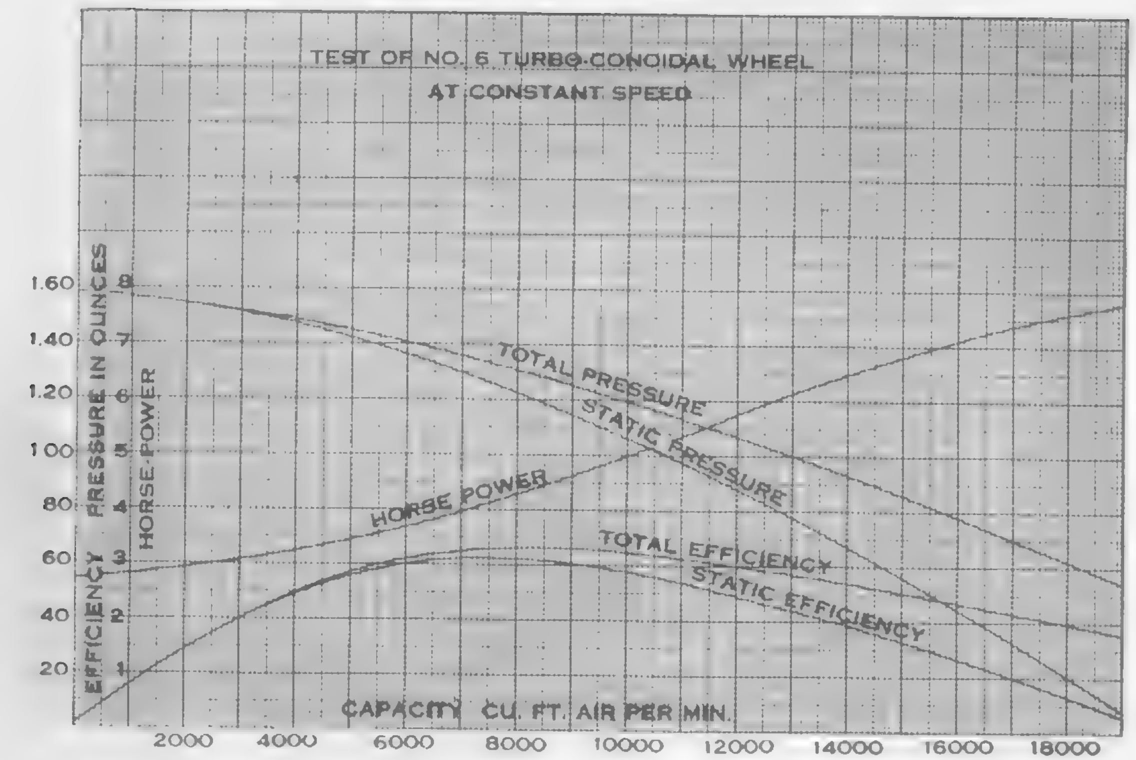
Size	Diameter of Blast Wheel Inches	Area of Outlet Square Ft.	2" Static Pressure = 1.154 Ounces R.P.M.	Volume Cubic Ft. per Min.	H.P.	2½" Static Pressure = 1.442 Ounces R.P.M.	Volume Cubic Ft. per Min.	H.P.	3" Static Pressure = 1.731 Ounces R.P.M.	Volume Cubic Ft. per Min.	H.P.	3½" Static Pressure = 2.019 Ounces R.P.M.	Volume Cubic Ft. per Min.	H.P.
2½	14¼	0.91	2,225	2,455	1.59	2,490	2,750	2.22	2,740	3,010	2.94	2,958	3,250	3.69
3	17¼	1.31	1,860	3,540	2.29	2,075	3,960	3.19	2,282	4,330	4.23	2,463	4,680	5.30
3½	20	1.79	1,595	4,800	3.12	1,780	5,390	4.35	1,958	5,890	5.75	2,115	6,360	7.22
4	22¾	2.33	1,395	6,270	4.08	1,559	7,050	5.68	1,713	7,700	7.52	1,850	8,320	9.45
4½	25¾	2.95	1,240	7,950	5.16	1,385	8,920	7.19	1,522	9,740	9.52	1,645	10,550	11.95
5	28½	3.64	1,117	9,800	6.37	1,249	11,000	8.87	1,370	12,000	11.75	1,480	13,000	14.75
5½	31½	4.41	1,015	11,880	7.72	1,133	13,300	10.75	1,245	14,550	14.23	1,345	15,750	17.85
6	34¼	5.25	932	14,120	9.18	1,040	15,800	12.78	1,141	17,300	16.92	1,232	18,700	21.25
6½	36¾	6.16	860	16,600	10.76	960	18,600	15.00	1,054	20,300	19.85	1,139	21,950	24.90
7	39¾	7.14	799	19,250	12.50	891	21,550	17.40	978	23,550	23.05	1,056	25,450	28.90
7½	42½	8.19	745	22,100	14.32	831	24,750	19.95	914	27,050	26.40	987	29,200	33.20
8	45½	9.33	700	25,100	16.30	780	28,150	22.70	856	30,800	30.10	925	33,300	37.75
8½	48	10.53	657	28,100	18.40	736	31,800	25.60	807	34,750	33.95	870	37,550	42.25
9	51¼	11.81	621	31,800	20.65	693	35,600	28.75	761	38,950	38.05	822	41,050	47.80
10	56¾	14.58	559	39,300	25.50	625	44,000	35.50	685	48,100	47.00	740	52,000	59.00
11	62½	17.64	507	47,450	30.85	567	53,250	42.95	623	58,150	56.90	673	62,900	71.45
12	68	21.00	465	56,500	36.75	520	63,500	51.10	570	69,250	67.70	616	74,950	85.00
13	73½	24.65	430	66,200	43.05	480	74,400	60.00	527	81,300	79.40	569	87,900	99.60
14	79	28.68	399	76,800	50.00	445	86,300	69.55	489	94,300	92.10	528	101,900	115.60
15	84¾	32.80	372	88,500	57.40	415	99,000	80.00	456	108,000	105.70	493	117,000	133.00
16	90¼	37.32	349	100,500	65.30	390	112,500	90.90	428	123,000	120.50	462	133,000	151.00

Total Pressure is 122.7% of the Rated Static Pressure

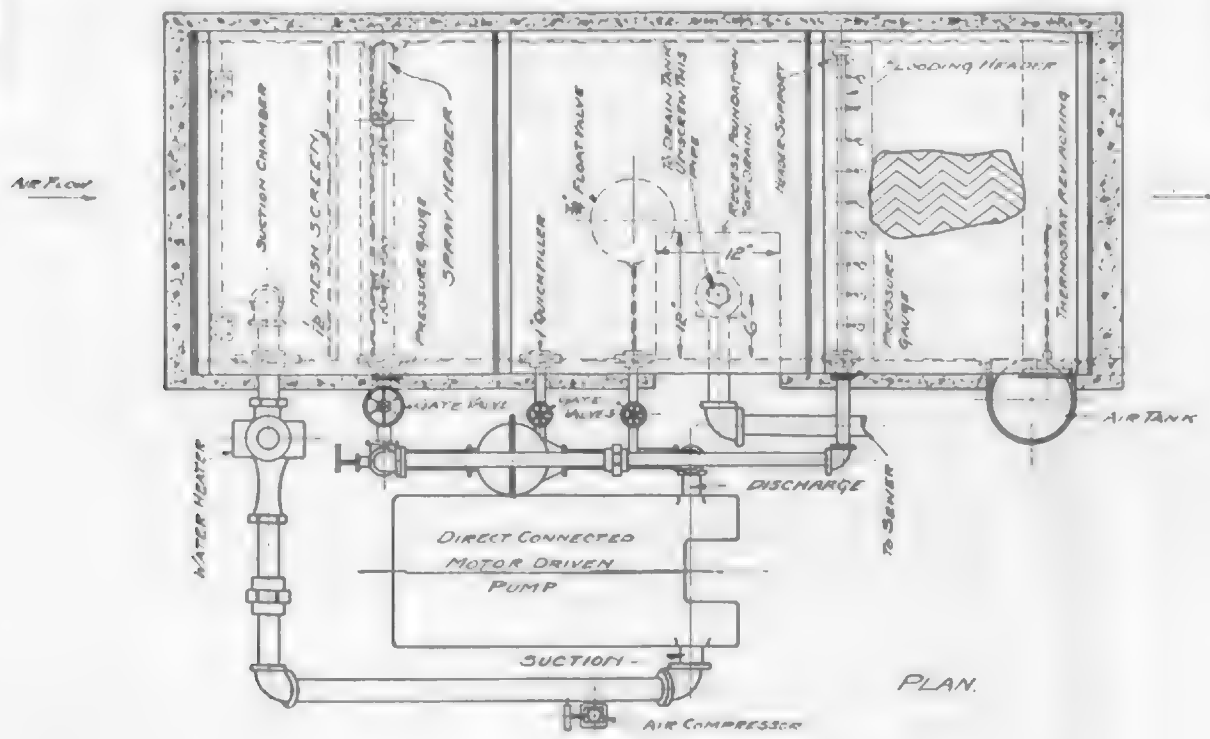


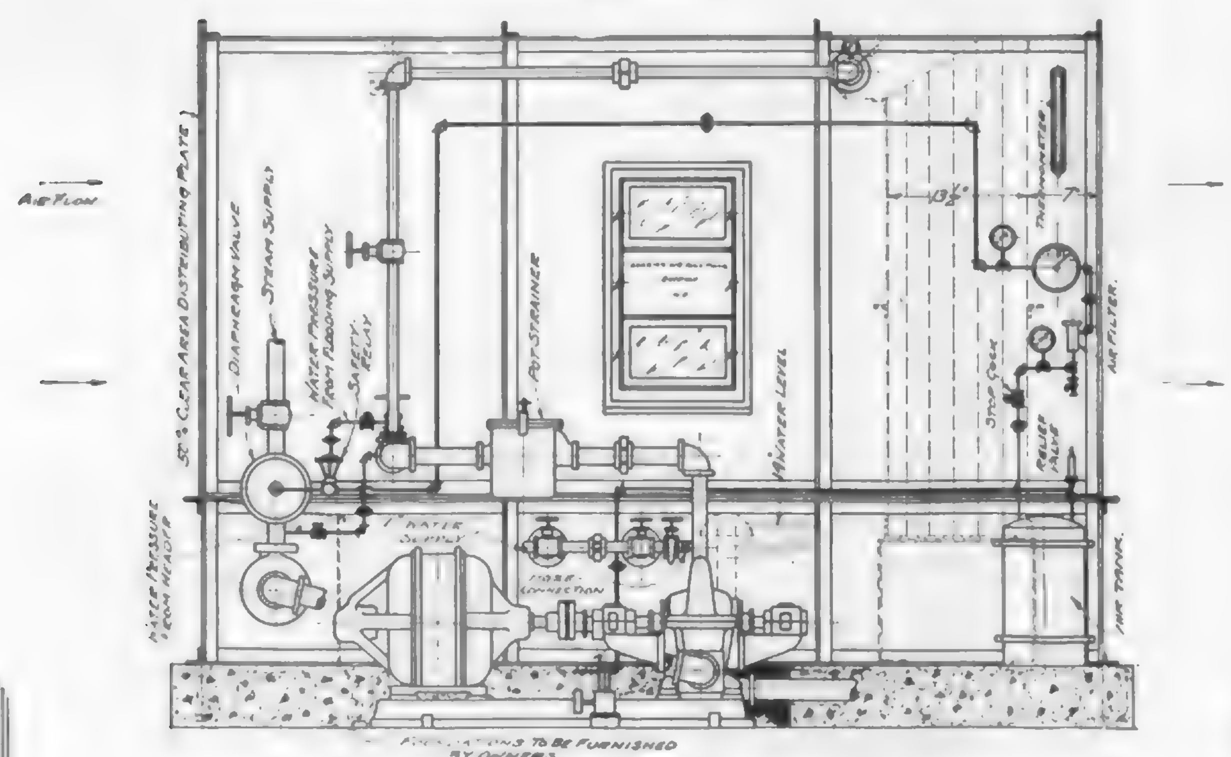


Fan System of Heating, Ventilating and Humidifying



Carrier Type "A" Air Washer with Humidity Control





ELEVATION

Carrier Type "A" Air Washer

Dimensions and Capacities

Square Feet Free Area	Square Feet Washing Surface	Size Door	Gallons per Minute: Spray	Flooding	Both	Water Pipes: To Pump	Fresh	Pump: Size	R P M	H. P.: Brake	Size Motor	Steam Pipe: 0 Pounds	5 Pounds	Height	Width	Length	Capacity Cubic Feet Air per Minute	Number
2.95	28.6	15½" x 26"	9	5	14	1½	¾	1½	1700	.9	2	1	1	4'—11½"	1'—5¼"	7'—2¾"	1500	1A
6.23	60.5		18	10	28	"	"	"	"	1.2	"	1½	1¼		2'—9"		3100	2A
9.50	92.5		27	15	42	2	"	"	"	1.5	"	2	1½		4'—0¾"		4800	3A
12.8	124		36	20	56	"	"	"	"	1.8	3	"	2		5'—4½"		6400	4A
16.0	155		45	25	70	"	"	"	"	2.1	"	2½	"		6'—8"		8000	5A
19.3	187		54	31	85	3	"	2	"	2.4	"	"	2½		7'—11¾"		9700	6A
22.6	219		63	36	99	"	"	"	"	2.7	"	3	"		9'—3½"		11300	7A
25.9	251		72	41	113	"	"	"	"	3.0	5	"	"		10'—7¼"		13000	8A
29.1	282		81	46	127	"	1"	2½	"	3.3	"	"	3		11'—11"		14600	9A
32.4	315		90	52	142	"	"	"	"	3.6	"	3½	"		13'—2¾"		16200	10A
35.7	346		99	57	156	"	"	"	"	3.9	"	"	"		14'—6¼"		17800	11A
4.13	39.2	15½" x 26"	11	5	16	2"	¾	1½	1700	1.0	2	1¼	1	5'—11½"	1'—5¼"	7'—2¾"	2100	1B
8.71	84.5		22	10	32	"	"	"	"	1.3	"	2	1½		2'—9"		4400	2B
13.3	129		33	15	48	"	"	"	"	1.6	"	"	"		4'—0¾"		6700	3B
17.9	174		43	20	63	"	"	"	"	1.9	3	2½	2½		5'—4½"		9000	4B
22.5	218		54	25	79	2½	"	2	"	2.3	"	3	"		6'—8"		11300	5B
27.1	263		65	31	96	3	"	"	"	2.6	"	"	3		7'—11¾"		13600	6B
31.7	308		76	36	112	"	"	"	"	3.0	5	"	"		9'—3½"		15800	7B
36.2	351		87	41	128	"	"	2½	"	3.3	"	3½	"		10'—7¼"		18100	8B
40.8	396		97	46	143	"	1"	"	"	3.6	"	"	3½		11'—11"		20400	9B
45.4	440		108	52	160	"	"	"	"	4.0	"	4	"		13'—2¾"		22700	10B
50.0	485		119	57	176	"	"	"	"	4.4	"	"	"		14'—6¼"		25000	11B
54.6	530		130	62	192	"	"	"	"	4.7	7½	"	4		15'—10"		27300	12B
6.40	62.0	16" x 30"	18	5	23	1½	¾	1½	1700	1.1	2	1½	1¼	7'—11½"	1'—5¼"	7'—2¾"	3200	1C
13.7	133		36	10	46	2	"	"	"	1.6	"	2	2		2'—9"		6900	2C
20.9	203		54	15	69	"	"	"	"	2.1	3	2½	2½		4'—0¾"		10500	3C
28.1	273		72	20	92	"	"	2	"	2.6	"	3	3		5'—4½"		14100	4C
35.3	343		90	25	115	"	"	"	"	3.1	5	3½	"		6'—8"		17700	5C
42.5	413		108	31	139	3	"	2½	"	3.6	"	"	3½		7'—11¾"		21300	6C
49.7	483		126	36	162	"	"	"	"	4.1	"	4	"		9'—3½"		24900	7C
56.9	553		144	41	185	"	"	"	"	4.6	"	"	4		10'—7¼"		28500	8C
64.1	623		162	46	208	"	1	"	"	5.1	7½	4½	"		11'—11"		32100	9C
71.3	693		180	52	232	4	"	3	"	5.6	"	5	4½		13'—2¾"		35700	10C
78.5	763		198	57	255	"	"	"	"	6.1	"	"	"		14'—6¼"		39300	11C
85.7	832		216	62	278	"	"	"	"	6.5	10	"	"		15'—10"		42800	12C
92.9	902		234	67	301	5	"	4	1120	7.1	"	6	5		17'—1¾"		46500	13C
100	970		252	73	325	"	"	"	"	7.5	"	"	"		18'—5½"		50000	14C
107	1040		270	78	348	"	"	"	"	7.9	"	"	"		19'—9¼"		53500	15C
114	1110		288	83	371	"	1½	"	"	8.3	"	"	6		21'—1"		57000	16C
122	1180		306	88	394	"	"	"	"	8.7	"	"	"		22'—4½"		61000	17C
129	1250		324	93	417	"	"	"	"	9.0	"	7	"		23'—8¼"		65000	18C
136	1320		342	99	441	"	"	"	"	9.4	"	"	"		25'—0"		68000	19C
143	1390		360	104	464	"	"	"	"	9.8	15	"	"		26'—3¾"		72000	20C
150	1460		378	109	487	"	"	"	"	10.1	"	"	"		27'—7½"		75000	21C
158	1530		396	114	510	"	"	"	"	10.5	"	"	7		28'—11"		79000	22C
165	1600		414	120	534	6	"	5	"	10.9	"	"	"		30'—2¾"		83000	23C

Carrier Type "A" Air Washer

Dimensions and Capacities

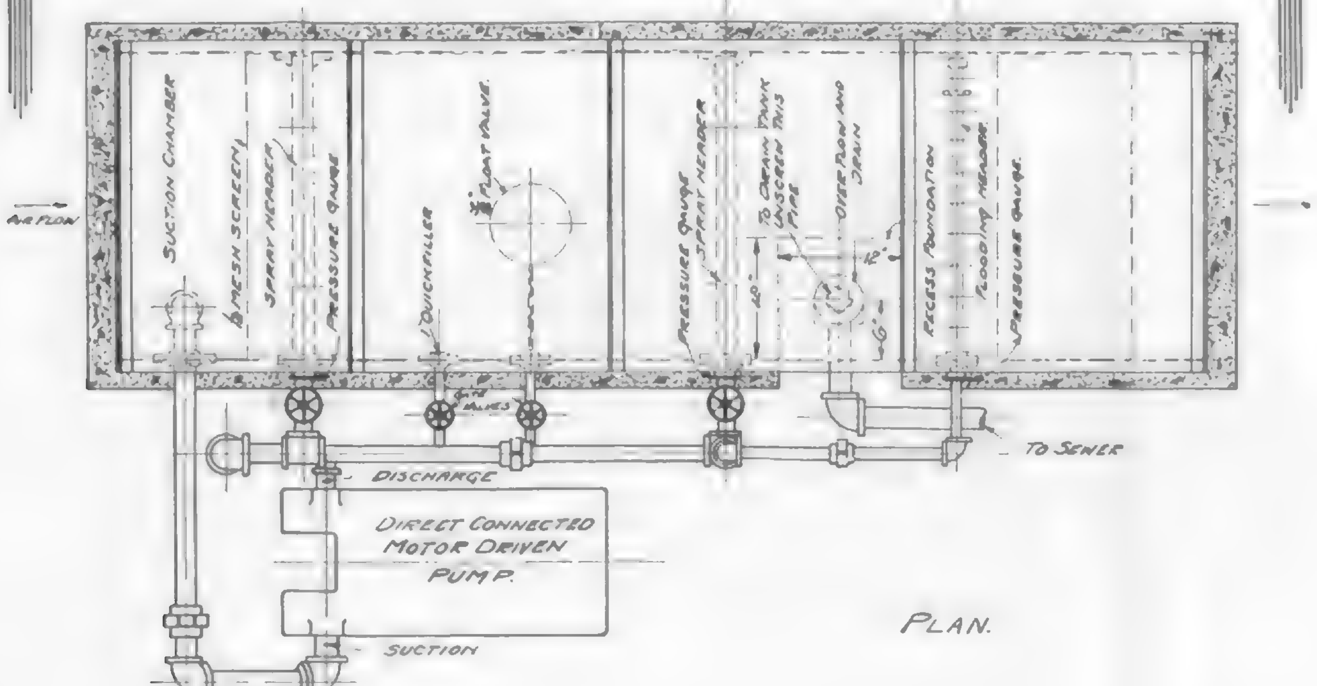
Square Feet Free Area	Square Feet Washing Surface	Size Door	Gallons per Minute: Spray	Flooding	Both	Water Pipes: To Pump	Fresh	Pump: Size	R. P. M.	H. P.: Brake	Size Motor	Steam Pipe: 0 Pounds	5 Pounds	Height	Width	Length	Capacity Cubic Feet Air per Minute	Number
8.9	86	16"x36"	24	5	29	1½	¾	1½	1700	1.2	3	2	1½	9'—1½"	1'—5¼"	7'—2⅛"	4400	1D
18.9	181	"	47	10	57	"	"	"	"	1.8	"	2½	2½	"	2'—9"	"	9400	2D
28.5	276	"	70	15	85	2½	"	2	"	2.5	"	3	3	"	4'—0½"	"	14300	3D
38.4	373	"	94	20	114	3	"	"	"	3.0	"	3½	"	"	5'—4"	"	19200	4D
47.9	468	"	117	25	142	"	"	2½	"	3.6	7½	"	3½	"	6'—8"	"	24100	5D
57.0	563	"	140	31	171	"	"	"	"	4.2	"	4½	4	"	7'—11⅞"	"	29000	6D
67.8	658	"	164	36	200	"	"	"	"	4.9	"	"	"	"	9'—3"	"	33900	7D
77.5	754	"	187	41	228	4	"	3	"	5.5	"	5	4½	"	10'—7"	"	38800	8D
87.4	847	"	210	46	256	"	1"	"	"	6.1	10	"	"	"	11'—11"	"	43700	9D
97.2	940	"	234	52	286	"	"	"	"	6.7	"	6	"	"	13'—2"	"	48600	10D
107	1040	"	258	57	315	5	"	4	1120	7.2	"	"	"	"	14'—6"	"	53500	11D
117	1140	"	281	62	343	"	"	"	"	7.8	"	"	6	"	15'—10"	"	58300	12D
127	1230	"	304	67	371	"	"	"	"	8.3	"	"	"	"	17'—1"	"	63000	13D
137	1330	"	328	73	401	"	"	"	"	8.7	15	7	"	"	18'—5"	"	68000	14D
146	1420	"	352	78	430	6	"	5	"	9.2	"	"	"	"	19'—9"	"	73000	15D
156	1520	"	375	83	458	"	1½	"	"	9.7	"	"	7	"	21'—1"	"	78000	16D
166	1610	"	398	88	486	"	"	"	"	10.1	"	"	"	"	22'—4"	"	83000	17D
176	1710	"	422	93	515	"	"	"	"	10.6	"	8	"	"	23'—8"	"	88000	18D
186	1800	"	445	99	544	"	"	"	"	11.0	"	"	"	"	25'—0"	"	92000	19D
195	1900	"	468	104	572	"	"	"	"	11.5	"	"	"	"	26'—4"	"	98000	20D
205	1990	"	492	109	601	"	"	"	"	11.9	"	"	8	"	27'—7"	"	103000	21D
215	2090	"	515	114	629	"	"	"	"	12.4	"	"	"	"	28'—11"	"	108000	22D
225	2190	"	538	120	658	"	"	"	"	12.9	"	"	"	"	30'—2"	"	113000	23D
11.2	109	16"x36"	29	5	34	1½	¾	1½	1700	1.3	3	2	2	11'—1⅜"	1'—5¼"	7'—2⅛"	5600	1E
23.5	229	"	58	10	68	2½	"	"	"	2.0	3	3	2½	"	2'—9"	"	11800	2E
36.1	350	"	87	15	102	3	"	2	"	2.8	5	3½	3	"	4'—0½"	"	18100	3E
48.6	472	"	115	20	135	"	"	2½	"	3.5	"	4	3½	"	5'—4"	"	24300	4E
61.0	592	"	144	25	169	"	"	"	"	4.2	"	4½	4	"	6'—8"	"	31000	5E
73.4	712	"	173	31	204	"	"	"	"	5.0	7½	5	4½	"	7'—11⅞"	"	36500	6E
85.8	833	"	202	36	238	4	"	3	"	5.7	"	"	"	"	9'—3"	"	42000	7E
98.2	953	"	230	41	271	"	"	"	"	6.4	"	6	5	"	10'—7"	"	49000	8E
110	1070	"	259	46	305	5	1"	4	1120	7.1	10	"	6	"	11'—11"	"	55000	9E
123	1190	"	288	52	340	"	"	"	"	7.8	"	"	"	"	13'—2"	"	62000	10E
135	1310	"	317	57	374	"	"	"	"	8.3	"	7	"	"	14'—6"	"	68000	11E
148	1430	"	346	62	408	"	"	"	"	8.9	"	"	"	"	15'—10"	"	74000	12E
160	1550	"	375	67	442	"	"	"	"	9.4	"	"	7	"	17'—1"	"	80000	13E
173	1680	"	404	73	477	"	"	"	"	10.0	15	"	"	"	18'—5"	"	87000	14E
185	1800	"	432	78	510	"	"	"	"	10.5	"	8	"	"	19'—9"	"	93000	15E
198	1920	"	462	83	545	6	1½	5	"	11.0	"	"	"	"	21'—1"	"	99000	16E
210	2040	"	490	88	578	"	"	"	"	11.5	"	"	8	"	22'—4"	"	105000	17E
222	2160	"	518	93	611	"	"	"	"	12.1	"	"	"	"	23'—8"	"	111000	18E
235	2280	"	547	99	646	"	"	"	"	12.6	"	10	"	"	25'—0"	"	118000	19E
247	2400	"	576	104	680	"	"	"	"	13.2	"	"	"	"	26'—4"	"	124000	20E
260	2520	"	605	109	714	"	"	"	"	13.7	"	"	"	"	27'—7"	"	130000	21E
273	2650	"	634	114	748	"	"	"	"	14.2	"	"	10	"	28'—11"	"	137000	22E
285	2770	"	662	120	782	"	"	"	"	14.7	20	"	"	"	30'—2"	"	143000	23E

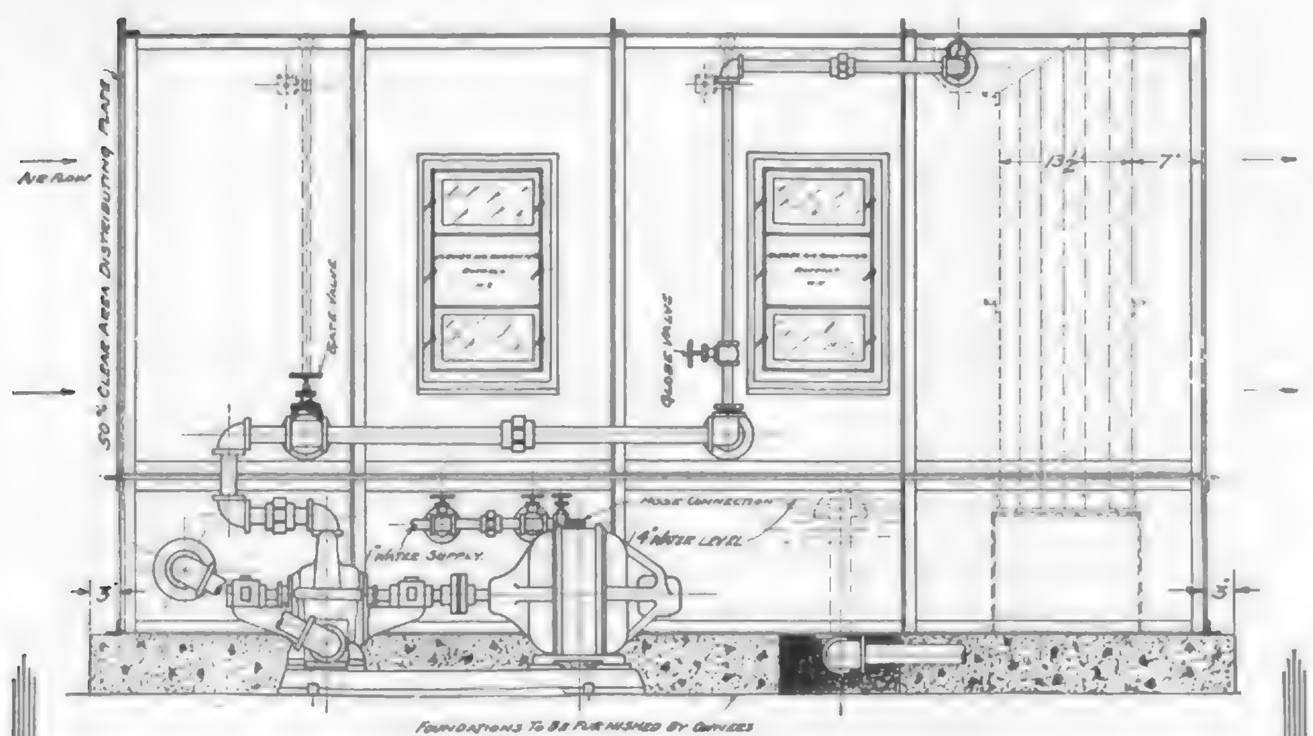
Carrier Type "A" Air Washer

Dimensions and Capacities

Square Feet Free Area	Square Feet Washing Surface	Size Door	Gallons per Minute: Spray	Flooding	Both	Water Pipes: To Pump	Fresh	Pump: Size	R. P. M.	H. P.: Brake	Size Motor	Steam Pipe: 0 Pounds	5 Pounds	Height	Width	Length	Capacity Cubic Feet Air per Minute	Number
13.5	131	16"x36"	36	5	41	2	¾	1½	1700	1.5	3	2	2	13'—1½"	1'—5¼"	7'—2⅛"	6800	1F
28.5	278	"	72	10	82	2½	"	"	"	2.3	"	3	3	"	2'—9"	"	14300	2F
43.7	424	"	108	15	123	3	"	2	"	3.2	5	4½	3½	"	4'—0½"	"	21900	3F
57.8	570	"	144	20	164	"	"	2½	"	4.1	"	4½	4	"	5'—4"	"	29200	4F
72.9	716	"	180	25	205	"	"	"	"	5.0	7½	"	4½	"	6'—8"	"	37000	5F
87.0	861	"	216	31	247	4	"	3	"	5.9	"	5	5	"	7'—11⅞"	"	44500	6F
101	1010	"	252	36	288	"	"	"	"	6.8	"	6	"	"	9'—3"	"	52000	7F
116	1160	"	288	41	329	5	"	4	1120	7.5	10	"	6	"	10'—7"	"	60000	8F
131	1300	"	324	46	370	"	1"	"	"	8.2	"	7	"	"	11'—11"	"	67000	9F
146	1440	"	360	52	412	"	"	"	"	8.9	"	"	"	"	13'—2"	"	75000	10F
161	1590	"	396	57	453	6	"	5	"	9.6	15	"	7	"	14'—6"	"	82000	11F
175	1740	"	432	62	494	"	"	"	"	10.3	"	8	"	"	15'—10"	"	90000	12F
191	1880	"	468	67	535	"	"	"	"	10.9	"	"	"	"	17'—1"	"	95000	13F
205	2030	"	504	73	577	"	"	"	"	11.5	"	"	8	"	18'—5"	"	105000	14F
221	2170	"	540	78	618	"	"	"	"	12.2	"	"	"	"	19'—9"	"	112000	15F
235	2320	"	576	83	659	"	1½	"	"	12.9	"	10	"	"	21'—1"	"	120000	16F
250	2470	"	612	88	700	"	"	"	"	13.5	"	"	"	"	22'—4"	"	127000	17F
265	2610	"	648	93	741	"	"	"	"	14.1	"	"	"	"	23'—8"	"	135000	18F
280	2760	"	684	99	783	"	"	"	"	14.7	20	"	10	"	25'—0"	"	142000	19F
295	2900	"	720	104	824	8	"	6	"	15.2	"	"	"	"	26'—4"	"	150000	20F
310	3050	"	756	109	865	"	"	"	"	15.7	"	"	"	"	27'—7"	"	157000	21F
325	3190	"	792	114	906	"	"	"	"	16.2	"	"	"	"	28'—11"	"	165000	22F
340	3340	"	828	120	948	"	"	"	"	16.6	"	"	"	"	30'—2"	"	172000	23F
15.9	154	16" x 36"	42	5	47	2	¾	1½	1700	1.6	3	2½	2	15'—2"	1'—5¼"	7'—2⅛"	8000	1G
33.6	326	"	83	10	93	3	"	2	"	2.6	3	3½	3	"	2'—9"	"	16800	2G
51.3	498	"	125	15	140	"	"	2½	"	3.6	5	4	3½	"	4'—0½"	"	25700	3G
69.0	670	"	166	20	186	"	"	"	"	4.6	"	4½	4	"	5'—4"	"	34500	4G
86.7	842	"	207	25	232	4	"	3	"	5.6	7½	5	4½	"	6'—8"	"	43300	5G
104	1010	"	249	31	280	"	"	"	"	6.6	"	6	5	"	7'—11⅞"	"	52000	6G
122	1180	"	291	36	326	5	"	4	1120	7.5	10	"	6	"	9'—3"	"	61000	7G
139	1360	"	332	41	373	"	"	"	"	8.3	"	7	"	"	10'—7"	"	70000	8G
157	1520	"	373	46	419	"	1	"	"	9.0	"	"	7	"	11'—11"	"	79000	9G
175	1700	"	414	52	466	"	"	"	"	9.8	15	8	"	"	13'—2"	"	88000	10G
193	1870	"	456	57	513	6	"	5	"	10.6	"	"	"	"	14'—6"	"	97000	11G
211	2040	"	497	62	559	"	"	"	"	11.3	"	"	8	"	15'—10"	"	106000	12G
228	2210	"	539	67	606	"	"	"	"	12.0	"	"	"	"	17'—1"	"	114000	13G
246	2380	"	580	73	653	"	"	"	"	12.8	"	10	"	"	18'—5"	"	123000	14G
264	2550	"	622	78	700	"	"	"	"	13.5	"	"	"	"	19'—9"	"	132000	15G
281	2730	"	663	83	746	"	1½	"	"	14.2	20	"	10	"	21'—1"	"	141000	16G
299	2890	"	704	88	792	"	"	"	"	14.8	"	"	"	"	22'—4"	"	150000	17G
316	3070	"	745	93	838	8	"	6	"	15.4	"	"	"	"	23'—8"	"	158000	18G
334	3240	"	787	99	886	"	"	"	"	15.9	"	"	"	"	25'—0"	"	167000	19G
352	3410	"	828	104	932	"	"	"	"	16.6	"	12	"	"	26'—4"	"	176000	20G
369	3580	"	870	109	979	"	"	"	"	17.2	"	"	"	"	27'—7"	"	185000	21G
387	3750	"	912	114	1026	"	"	"	"	17.8	"	"	"	"	28'—11"	"	194000	22G
405	3920	"	953	120	1073	"	"	"	"	18.5	25	"	"	"	30'—2"	"	203000	23G

Carrier Type "B" Air Washer with Humidity Control





Fan System of Heating, Ventilating and Humidifying

Carrier Type "B" Air Washer

Dimensions and Capacities

Square Feet Free Area	Square Feet Washing Surface	Size Door	Gallons per Minute: Spray	Flooding	Both	Water Pipes: To Pump	Fresh	Pump: Size	R. P. M.	H. P.: Brake	Size Motor	Steam Pipe: 0 Pounds	5 Pounds	Height	Width	Length	Capacity Cubic Feet Air per Minute	Number
2.95	28.6	15½"×26"	18	5	23	1½	¾	1½	1700	1.1	2	1	1	4'—1½"	1'—5¼	9'—0⅝"	1500	1A
6.24	60.5		36	10	46	2	"	"	"	1.6	"	1½	1¼		2'—9"		3100	2A
9.50	92.5		54	15	69	"	"	"	"	2.1	3	2	1½		4'—0¾		4800	3A
12.8	124		72	20	92	3	"	2"	"	2.6	"	"	2		5'—4½		6400	4A
16.0	155		90	25	115	"	"	"	"	3.0	5	2½	"		6'—8"		8000	5A
19.3	187		108	31	139	"	1	2½	"	3.6	"	"	2½		7'—11¾		9700	6A
22.6	219		126	36	162	"	"	"	"	4.1	"	3	"		9'—3½		11300	7A
25.9	251		144	41	185	"	"	"	"	4.5	"	"	"		10'—7¼		13000	8A
29.1	282		162	46	208	"	"	"	"	5.1	7½	"	3		11'—11"		14600	9A
32.4	315		180	52	232	4	"	3"	"	5.6	"	3½	"		13'—2¾		16200	10A
35.7	346		198	57	255	"	1¼	"	"	5.8	"	"	"		14'—6¼		17800	11A
4.43	39.2	15½"×26"	22	5	27	1½	¾	1½	1700	1.2	2	1¼	1	5'—1½"	1'—5¼	9'—0⅝"	2100	1B
8.71	84.5		44	10	54	2	"	"	"	1.7	3	2	1½		2'—9"		4400	2B
13.3	129		66	15	81	2½	"	2	"	2.3	"	"	2		4'—0¾		6700	3B
17.9	174		86	20	106	3	"	"	"	2.9	5	2½	2½		5'—4½		9000	4B
22.5	218		108	25	133	"	"	2½	"	3.4	"	3	"		6'—8"		11300	5B
27.1	263		130	31	161	"	1	"	"	4.0	"	"	3		7'—11¾		13600	6B
31.7	308		152	36	188	"	"	"	"	4.6	"	"	"		9'—3½		15800	7B
36.2	351		174	41	215	4	"	3	"	5.2	7½	3½	"		10'—7¼		18100	8B
40.8	396		194	46	240	"	"	"	"	5.7	"	"	3½		11'—11"		20400	9B
45.4	440		216	52	268	"	"	"	"	6.3	"	4	"		13'—2¾		22700	10B
50.0	485		238	57	295	5	1¼	4	"	6.9	"	"	"		14'—6¼		25000	11B
54.6	530		260	62	322	"	"	"	1120	7.4	10	"	4		15'—10"		27300	12B
6.40	62.0	16"×36"	36	5	41	2	¾	1½	1700	1.5	2	1½	1¼	7'—1¼"	1'—5¼	9'—0½"	3200	1C
13.7	133		72	10	82	2½	"	2	"	2.3	3	2	2		2'—9"		6900	2C
20.9	203		108	15	123	3	"	"	"	3.2	5	2½	2½		4'—0¾		10500	3C
28.1	273		144	20	164	"	"	2½	"	4.1	"	3	3		5'—4½		14100	4C
35.3	343		180	25	205	"	"	"	"	5.0	7½	3½	"		6'—8"		17700	5C
42.5	413		216	31	247	4	1"	3	"	5.9	"	"	3½		7'—11¾		21300	6C
49.7	483		252	36	288	"	"	"	"	6.7	"	4	"		9'—3½		24900	7C
56.9	553		288	41	329	5	"	4	1120	7.6	10	"	4		10'—7¼		28500	8C
64.1	623		324	46	370	"	"	"	"	8.2	"	4½	"		11'—11"		32100	9C
71.3	693		360	52	412	"	"	"	"	8.8	"	5	4½		13'—2¾		35700	10C
78.5	763		396	57	453	"	1¼	"	"	9.6	15	"	"		14'—6¼		39300	11C
85.7	832		432	62	494	"	"	"	"	10.2	"	"	"		15'—10"		42800	12C
92.9	902		478	67	545	6	"	5	"	11.1	"	6	5		17'—1½		46500	13C
100	970		504	73	577	"	"	"	"	11.6	"	"	"		18'—5¼		50000	14C
107	1040		540	78	618	"	"	"	"	12.2	"	"	"		19'—9¼		53500	15C
114	1110		576	83	659	"	"	"	"	12.9	"	"	6		21'—1"		57000	16C
122	1180		612	88	700	"	1½	"	"	13.5	"	"	"		22'—4½		61000	17C
129	1250		648	93	741	"	"	"	"	14.0	"	7	"		23'—8¼		65000	18C
136	1320		684	99	783	"	"	"	"	14.7	20	"	"		25'—0"		68000	19C
143	1390		720	104	824	8	"	6	"	15.2	"	"	"		26'—3¾		72000	20C
150	1460		756	109	865	"	"	"	"	15.7	"	"	"		27'—7½		75000	21C
158	1530		792	114	906	"	"	"	"	16.2	"	"	7		28'—11"		79000	22C
165	1600		828	120	1008	"	2	"	"	17.6	"	"	"		30'—2¾		83000	23C

Carrier Type "B" Air Washer

Dimensions and Capacities

Square Feet Free Area	Square Feet Washing Surface	Size Door	Gallons per Minute: Spray	Flooding	Both	Water Pipes: To Pump	Fresh	Pump: Size	R. P. M.	H. P.: Brake	Size Motor	Steam Pipe: 0 Pounds	5 Pounds	Height	Width	Length	Capacity Cubic Feet Air per Minute	Number
8.85	86.0	16" x 36"	48	5	53	2	¾	1½	1700	1.7	3	2	1½	9'—1⅝"	1'—5¼"	9'—0¼"	4400	1D
18.7	181		94	10	104	3	"	2	"	2.8	5	2¼	2½		2'—9"		9100	2D
28.5	276		140	15	155	"	"	2½	"	3.9	"	3	3		4'—0¾"		13900	3D
38.4	373		188	20	208	"	"	"	"	5.0	7½	3½	"		5'—4½"		19200	4D
48.2	468		244	25	269	4	"	3	"	6.1	"	"	3½		6'—8"		24300	5D
58.0	563		280	31	311	5	1	4	1120	7.3	10	4½	4		7'—11¾"		29000	6D
67.8	658		328	36	364	"	"	"	"	8.2	"	"	"		9'—3¼"		33900	7D
77.6	751		371	41	415	"	"	"	"	9.0	"	5	4½		10'—7¼"		38800	8D
87.4	847		423	46	466	"	"	"	"	9.8	15	"	5		11'—11"		43700	9D
97.2	940		468	52	520	"	"	"	"	10.7	"	6	"		13'—2½"		48600	10D
107	1040		516	57	573	6	1¼	5	"	11.5	"	"	"		14'—6¼"		53500	11D
117	1140		562	62	624	"	"	"	"	12.3	"	"	6		15'—10"		59000	12D
127	1230		608	67	675	"	"	"	"	13.1	"	"	"		17'—1¾"		63000	13D
137	1330		656	73	729	"	"	"	"	13.9	"	7	"		18'—5½"		68000	14D
146	1420		704	78	782	"	"	"	"	14.6	20	"	"		19'—9¼"		73000	15D
156	1520		750	83	833	8	"	6	"	15.3	"	"	7		21'—1"		78000	16D
166	1610		796	88	884	"	1½	"	"	15.9	"	"	"		22'—4½"		83000	17D
176	1710		844	93	937	"	"	"	"	16.6	"	8	"		23'—8¼"		88000	18D
186	1800		890	99	989	"	"	"	"	17.3	"	"	"		25'—0"		93000	19D
195	1900		936	104	1040	"	"	"	"	18.0	"	"	"		26'—3¾"		98000	20D
205	1990		984	109	1093	"	"	"	"	18.7	25	"	8		27'—7½"		103000	21D
215	2090		1020	114	1134	"	"	"	"	19.3	"	"	"		28'—11"		108000	22D
225	2180		1076	120	1196	"	2	"	"	20.2	"	"	"		30'—2"		113000	23D
11.2	109	16" x 36"	58	5	63	2	¾	1½	1700	1.9	3	2	2	11'—1⅝"	1'—5¾"	9'—0¼"	5600	1E
23.6	229		116	10	126	3	"	2½	"	3.3	5	3	2½		2'—9½"		11800	2E
36.1	350		174	15	189	"	"	"	"	4.6	"	3½	3		4'—1¼"		18100	3E
48.6	472		230	20	250	4	"	3	"	5.9	7½	4	3½		5'—5"		24500	4E
61.0	592		288	25	313	5	"	4	1200	7.2	10	4½	4		6'—8¾"		31000	5E
73.4	712		346	31	377	"	1	"	"	8.4	"	5	4½		8'—0½"		36700	6E
85.8	833		404	36	440	"	"	"	"	9.3	"	"	"		9'—4"		42900	7E
98.2	953		460	41	501	"	"	"	"	10.3	15	6	5		10'—7¾"		49100	8E
110	1070		518	46	564	6	"	5	"	11.4	"	"	6		11'—11½"		55000	9E
123	1190		576	52	628	"	"	"	"	12.3	"	"	"		13'—3"		62000	10E
135	1310		634	57	691	"	1¼	"	"	13.3	"	7	"		14'—6¾"		68000	11E
148	1430		692	62	754	"	"	"	"	14.3	"	"	"		15'—10½"		74000	12E
160	1550		750	67	817	8	"	6	"	15.2	20	"	7		17'—2¼"		80000	13E
173	1680		808	73	881	"	"	"	"	15.9	"	"	"		18'—6"		87000	14E
185	1800		864	78	942	"	"	"	"	16.8	"	8	"		19'—9¾"		93000	15E
198	1920		924	83	1007	"	"	"	"	17.6	"	"	"		21'—1½"		99000	16E
210	2040		980	88	1068	"	1½	"	"	18.4	25	"	8		22'—5"		105000	17E
222	2160		1036	93	1129	"	"	"	"	19.2	"	"	"		23'—8¾"		111000	18E
235	2280		1094	99	1193	"	"	"	"	20.2	"	10	"		25'—0½"		118000	19E
247	2400		1152	104	1256	"	"	7	"	21.2	"	"	"		26'—4¼"		124000	20E
260	2520		1210	109	1319	"	"	"	"	22.1	"	"	"		27'—8"		130000	21E
272	2650		1268	114	1382	"	"	"	"	23.3	30	"	10		28'—11¾"		137000	22E
285	2770		1326	120	1446	"	2	"	"	24.4	"	"	"		30'—3½"		143000	23E

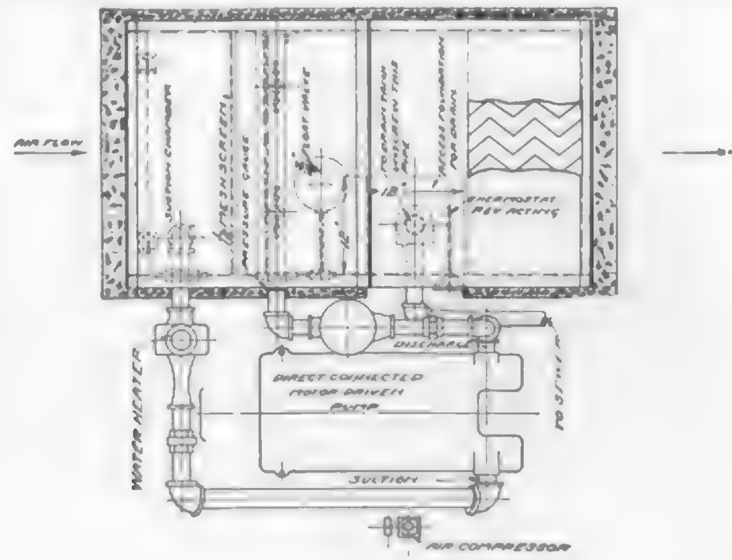
FAN SYSTEM OF HEATING, VENTILATING AND HUMIDIFYING

Carrier Type "B" Air Washer

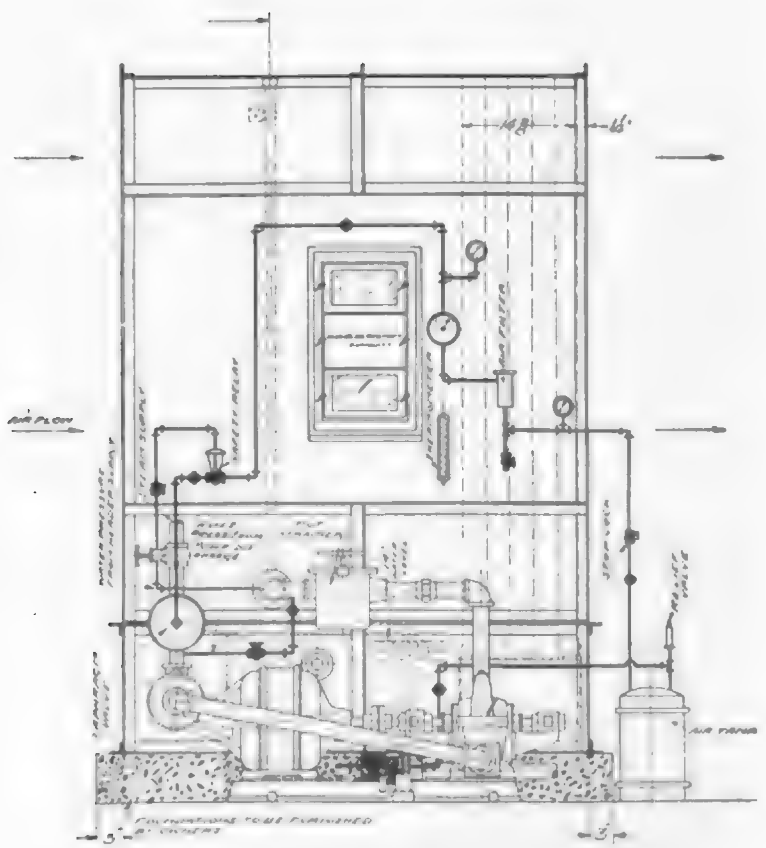
Dimensions and Capacities

Square Feet Free Area	Square Feet Washing Surface	Size Door	Gallons per Minute: Spray	Flooding	Both	Water Pipes: To Pump	Fresh	Pump: Size	R. P. M.	H. P.: Brake	Size Motor	Steam Pipe: 0 Pounds	5 Pounds	Height	Width	Length	Capacity Cubic Feet Air per Minute	Number
13.5	131	16" x 36"	72	5	77	2½	¾	2"	1700	2.2	3	2½	2	13'—1¾"	1'—5¾"	9'—0¼"	6800	1F
28.6	275		144	10	154	3	"	2½	"	3.9	5	3	3		2'—9½"		14300	2F
43.7	423		216	15	231	4	"	3	"	5.5	7½	3½	3½		4'—1¼"		22900	3F
58.8	570		288	20	308	5	"	4	1120	7.2	10	4½	4		5'—5"		29400	4F
73.9	716		360	25	385	"	"	"	"	8.5	"	5	4½		6'—8½"		37000	5F
89.0	864		432	31	463	"	1	"	"	9.7	15	"	5		8'—0¼"		44500	6F
104	1010		504	36	540	6	"	5	"	11.0	"	6	"		9'—4"		52000	7F
119	1160		576	41	617	"	"	"	"	12.2	"	"	6		10'—7¾"		60000	8F
134	1300		648	46	694	"	"	"	"	13.4	"	7	"		11'—11½"		67000	9F
149	1440		720	52	772	"	"	"	"	14.5	20	"	"		13'—3"		75000	10F
164	1590		792	57	849	7	1¼	6	"	15.5	"	"	7		14'—6¾"		82500	11F
179	1740		864	62	926	"	"	"	"	16.5	"	8	"		15'—10½"		90000	12F
194	1880		936	67	1003	"	"	"	"	17.5	"	"	"		17'—2¼"		97000	13F
209	2130		1008	73	1081	"	"	"	"	18.5	25	"	8		18'—6"		105000	14F
224	2270		1080	78	1158	"	"	"	"	19.6	"	"	"		19'—9¾"		112000	15F
239	2320		1152	83	1235	8	"	7	"	20.8	"	10	"		21'—1½"		120000	16F
254	2470		1224	88	1312	"	1½	"	"	22.1	"	"	"		22'—5"		127000	17F
269	2610		1296	93	1389	"	"	"	"	23.4	30	"	"		23'—8¾"		135000	18F
284	2760		1368	99	1467	"	"	"	"	24.7	"	"	10		25'—0½"		142000	19F
299	2900		1440	104	1544	"	"	"	"	26.1	"	"	"		26'—4¼"		150000	20F
314	3050		1512	109	1621	10"	"	8	"	27.4	35	"	"		27'—8"		157000	21F
329	3190		1584	114	1698	"	"	"	"	28.7	"	"	"		28'—11½"		165000	22F
344	3340		1656	120	1776	"	2	"	"	30.0	"	"	"		30'—3¼"		172000	23F
15.9	154	16" x 36"	84	5	89	2½	¾	2"	1700	2.5	3	2½	2	15'—2"	1'—6¾"	9'—0¼"	8000	1G
33.6	326		166	10	176	3	"	2½	"	4.4	5	3½	3		2'—10"		16800	2G
51.3	498		250	15	265	4	"	3	"	6.3	7½	4	3½		4'—1¾"		25700	3G
69.0	670		332	20	352	5	"	4	1120	8.0	10	4½	4		5'—5½"		34500	4G
86.7	842		414	25	439	"	"	"	"	9.4	"	5	4½		6'—9"		43400	5G
104	1010		498	31	529	6	1	5	"	10.8	15	6	5		8'—0¾"		52000	6G
122	1180		580	36	616	"	"	"	"	12.2	"	"	6		9'—4½"		61000	7G
140	1360		666	41	707	"	"	"	"	13.6	"	7	"		10'—8¼"		70000	8G
157	1520		746	46	792	"	"	"	"	14.8	20	"	7		12'—0"		79000	9G
175	1700		828	52	880	8	"	6	"	15.8	"	8	"		13'—3½"		88000	10G
193	1870		912	57	969	"	1¼	"	"	17.1	"	"	"		14'—7¼"		97000	11G
211	2050		994	62	1056	"	"	"	"	18.2	"	"	8		15'—11"		105000	12G
228	2210		1078	67	1145	"	"	"	"	19.5	25	"	"		17'—2¾"		114000	13G
246	2390		1160	73	1233	"	"	7	"	20.8	"	10	"		18'—6½"		123000	14G
263	2550		1244	78	1322	"	"	"	"	22.3	"	"	"		19'—10¼"		132000	15G
281	2730		1326	83	1409	"	"	"	"	23.8	30	"	10		21'—2"		141000	16G
298	2890		1408	88	1496	"	1½	"	"	25.3	"	"	"		22'—5½"		149000	17G
316	3070		1490	93	1583	"	"	"	"	26.7	"	"	"		23'—9¼"		158000	18G
334	3240		1574	99	1673	10	"	8	"	28.3	35	"	"		25'—1"		167000	19G
351	3410		1656	104	1760	"	"	"	"	29.8	"	12	"		26'—4¾"		176000	20G
369	3580		1740	109	1849	"	"	"	"	31.3	"	"	"		27'—8½"		185000	21G
388	3770		1824	114	1938	"	"	"	"	32.8	40	"	"		29'—0"		194000	22G
405	3930		1906	120	2026	"	2	"	"	34.4	"	"	"		30'—3¾"		[illegible]	23G

Carrier Type "C" Air Washer with Humidity Control



PLAN



ELEVATION

FAN SYSTEM OF HEATING, VENTILATING AND HUMIDIFYING

Carrier Type "C" Air Washer

Dimensions and Capacities

Square Feet Free Area	Square Feet Washing Surface	Size Door	Water Pipes: To Pump	Water Pipes: Fresh	Pump: Size	Pump: R. P. M.	Pump: H. P. Brake	Pump: H. P. Size Motor	Steam Pipe: 0 Pounds	Steam Pipe: 5 Pounds	Height	Width	Length	Capacity Cubic Feet Air per Minute	Number
3.34	11.0	15½"x23"	1½	¾	1½	1700	.775	½	1	1"	4'—11½"	1'—5¼	4'—10"	1700	1A
7.05	23.0		"	"	"	"	.95	"	1½	1½		2'—9"		3500	2A
10.8	35.2		"	"	"	"	1.15	"	2	2"		4'—0¾		5400	3A
14.5	47.2		"	"	"	"	1.35	"	2½	"		5'—4½		7300	4A
18.2	59.4		2	"	"	"	1.55	"	"	2½		6'—8"		9100	5A
21.9	71.4		"	"	"	"	1.72	3	"	"		7'—11¾		11000	6A
25.6	83.4		"	"	"	"	1.92	"	3	"		9'—3½		12800	7A
29.3	95.4		"	"	"	"	2.10	"	"	3"		10'—7¼		14600	8A
33.0	107.6		2½	"	2	"	2.30	"	3½	"		11'—11"		16500	9A
36.7	119.6		"	"	"	"	2.50	"	"	"		13'—2¼		18300	10A
40.5	132.0		3	"	"	"	2.70	5	"	3½		14'—6¼		20200	11A
4.52	14.8	15½"x26"	1½	¾	1½	1700	.80	2	1¼	1¼	5'—11½"	1'—5¼	4'—10"	2300	1B
9.54	31.0		"	"	"	"	1.05	"	2	1½		2'—9"		4800	2B
14.5	47.2		"	"	"	"	1.30	"	2½	2		4'—0¾		7300	3B
19.6	63.8		2	"	"	"	1.50	"	"	2½		5'—4½		9800	4B
24.6	80.0		"	"	"	"	1.72	3	3	"		6'—8"		12500	5B
29.6	96.4		"	"	"	"	1.97	"	"	3		7'—11¾		14800	6B
34.6	112.8		2½	"	2	"	2.20	"	3½	"		9'—3½		17300	7B
39.6	129.0		"	"	"	"	2.42	"	"	3½		10'—7¼		19800	8B
44.6	145.2		"	"	"	"	2.65	5	"	"		11'—11"		22300	9B
49.6	162.0		"	"	"	"	2.90	"	4	"		13'—2½		24800	10B
54.6	178.0		"	"	"	"	3.12	"	"	4		14'—6¼		27300	11B
59.6	194.0		3	"	2½	"	3.35	"	4½	"		15'—10"		29800	12B
6.88	22.4	16"x30"	1½	¾	1½	1700	.95	2	1½	1½	7'—11½"	1'—5¼	4'—10"	3400	1C
14.5	47.2		"	"	"	"	1.35	"	2	2		2'—9"		7300	2C
22.1	72.		2	"	"	"	1.72	3	2½	2½		4'—0¾		11000	3C
29.8	97.2		"	"	"	"	2.10	"	3	3		5'—4½		14900	4C
37.4	122		2½	"	2	"	2.50	"	3½	"		6'—8"		18700	5C
45.1	147		3	"	"	"	2.87	5	"	3½		7'—11¾		22500	6C
52.7	172		"	"	"	"	3.27	"	4	"		9'—3½		26300	7C
60.4	197		"	"	2½	"	3.65	"	4½	4		10'—7¼		30700	8C
68.0	222		"	"	"	"	4.05	"	"	"		11'—11"		34000	9C
75.5	246		"	"	"	"	4.43	"	5	4½		13'—2½		37800	10C
83.2	272		"	"	"	"	4.82	7½	"	"		14'—6¼		41600	11C
91.0	296		4	"	3	"	5.20	"	6	5		15'—10"		45500	12C
98.3	320		"	"	"	"	5.58	"	"	"		17'—1¾		49200	13C
106	346		"	"	"	"	5.97	"	"	"		18'—5½		53000	14C
114	372		"	"	"	"	6.45	"	"	6		19'—9¼		57000	15C
121	394		"	"	"	"	6.73	"	"	"		21'—1"		60500	16C
129	420		5	"	4	1120	7.10	10	"	"		22'—4½		64500	17C
136	444		"	"	"	"	7.50	"	"	"		23'—8¼		68000	18C
144	470		"	"	"	"	7.80	"	7	"		25'—0"		72000	19C
152	492		"	"	"	"	8.10	"	"	7		26'—3¾		76000	20C
159	518		"	"	"	"	8.47	"	"	"		27'—7½		80000	21C
167	544		"	"	"	"	8.65	"	"	"		28'—11"		84000	22C
175	570		"	"	"	"	8.95	"	"	"		30'—2¾		88000	23C

Carrier Type "C" Air Washer

Dimensions and Capacities

Square Feet Free Area	Square Feet Washing Surface	Size Door	Water Pipes: To Pump	Water Pipes: Fresh	Pump: Size	Pump: R. P. M.	Pump H. P.: Brake	Pump H. P.: Size Motor	Steam Pipe: 0 Pounds	Steam Pipe: 5 Pounds	Height	Width	Length	Capacity Cubic Feet Air per Minute	Number
9.24	30.2	16" x 36"	1½	¾	1½	1700	1.10	2	2	1½	9'—1½"	1'—5¼"	4'—10"	4600	1D
19.5	63.6						1.57		2½	2½		2'—9"		9800	2D
29.8	97.2		2½				2.07	3		3		4'—0½"		14900	3D
40.	130.4		3		2		2.57		3½	3½		5'—4¼"		20000	4D
50.3	164						3.07	5	4			6'—[illegible]		25200	5D
60.6	198				2½		3.57		4½	4		7'—11¼"		30300	6D
70.8	230						4.07			4½		9'—3¼"		35400	7D
81.0	264						4.57		5			10'—7¾"		40500	8D
91.3	298						5.07	7½	6	5		11'—11"		45700	9D
101	330		4		3		5.57					13'—3"		50500	10D
112	366						6.07			6		14'—6¾"		56000	11D
122	398						6.57					15'—10"		61000	12D
132	430		5		4	1120	7.07	10	7			17'—1¾"		66000	13D
142	464						7.57					18'—[illegible]		71000	14D
152	500						7.95			7		19'—9¼"		76000	15D
163	532						8.30					21'—1"		81000	16D
173	564						8.70		8			22'—4¼"		86000	17D
183	596						9.10					23'—8¼"		91000	18D
194	632						9.45	15				25'—0"		97000	19D
204	664						9.80					26'—3¼"		102000	20D
214	698						10.20			8		27'—7¾"		107000	21D
225	734						10.55					28'—11"		113000	22D
235	766		6		5		10.90					30'—2¼"		118000	23D
11.6	37.8	16" x 36"	1½	¾	1½	1700	1.20	2	2	2	11'—1¾"	1'—5¼"	4'—10"	5800	1E
24.7	80		2				1.82	3	3	2½		2'—9¾"		12300	2E
37.4	122		2½		2		2.42		3½	3		4'—1¼"		18700	3E
50.3	164						3.02	5	4	3½		5'—5"		25200	4E
63.1	206		3		2½		3.65		4½	4		6'—8¾"		31600	5E
76	248						4.27		5	4½		8'—0¼"		38000	6E
89	290						4.90	7½		5		9'—4"		44500	7E
102	332		4		3		5.50		6			10'—7¾"		51000	8E
115	376						6.12			6		11'—11¼"		57500	9E
127	414						6.72					13'—3"		63500	10E
140	456		5		4	1120	7.32	10	7	7		14'—6¾"		70000	11E
153	500						7.85					15'—10¼"		76500	12E
166	540						8.30					17'—2¼"		83000	13E
179	584						8.80		8			18'—6"		89500	14E
192	626						9.20					19'—9¾"		96000	15E
204	664						9.70	15				21'—1"		102000	16E
217	706						10.20			8		22'—5"		108000	17E
230	750						10.60					23'—8¾"		115000	18E
243	792		6		5		11.10		10			25'—0¼"		122000	19E
256	836						11.50					26'—4¼"		128000	20E
268	874						12.00					27'—8"		134000	21E
282	920						12.50					28'—11¾"		141000	22E
295	962						12.90			10		30'—3¼"		148000	23E

"Buffalo"

Carrier Type "C" Air Washer

Dimensions and Capacities

Square Feet Free Area	Square Feet Washing Surface	Size Door	Water Pipes: To Pump	Water Pipes: Fresh	Pump: Size	Pump: R. P. M.	Pump H. P.: Brake	Pump H. P.: Size Motor	Steam Pipe: 0 Pounds	Steam Pipe: 5 Pounds	Height	Width	Length	Capacity Cubic Feet Air per Minute	Number
14.	45.6	16" x 36"	1½	¾	1½	1700	1.45	2	2	2	13'—1¼"	1'—5¼"	4'—10"	7000	1F
29.5	96		2½				2.12	3	3	3		2'—9½"		14800	2F
45	146		3		2		2.90	5	3½	3½		4'—1¼"		22500	3F
60.5	198				2½		3.67		4½	4		5'—5"		30300	4F
76	248						4.42		5	4½		6'—8¾"		38000	5F
91.5	298		4		3		5.20	7½	6	5		8'—0¼"		45700	6F
107	350						5.97					9'—4"		53500	7F
122	398						6.72			6		10'—7¾"		61000	8F
138	450		5		4	1120	7.47	10	7			12'—11¼"		69000	9F
154	502						8.10			7		13'—3"		77000	10F
169	550						8.62		8			14'—6¾"		85000	11F
185	606						9.20					15'—10½"		93000	12F
200	652						9.80	15				17'—2¼"		100000	13F
216	704						10.40			8		18'—6"		108000	14F
232	756		6		5		10.95					19'—9¾"		116000	15F
246	804						11.50		10			21'—1½"		125000	16F
262	854						12.10					22'—5"		131000	17F
278	908						12.70			10		23'—8¾"		139000	18F
294	960						13.20					25'—1"		147000	19F
308	1004						13.75					26'—4¼"		154000	20F
324	1056						14.25					27'—8"		162000	21F
340	1108						14.75	20				28'—11"		170000	22F
356	1160		8		6		15.25					30'—3¼"		178000	23F
16.4	53.4	16" x 36"	2	¾	1½	1700	1.47	2	2½	2	15'—2"	1'—6¼"	4'—10"	8200	1G
34.5	112		2½		2		2.35	3	3½	3		2'—10"		17300	2G
52.6	172		3				3.25	5	4	3½		4'—1¼"		26300	3G
70.7	230				2½		4.12		4½	4½		5'—5½"		35300	4G
89	290						5.00	7½	5	5		6'—9"		44500	5G
107	350		4		3		5.90		6			8'—0¾"		53500	6G
125	408						6.77			6		9'—4¼"		62500	7G
144	470		5		4	1120	7.60	10	7			10'—8¼"		72000	8G
162	528						8.30			7		12'—0"		81000	9G
180	586						8.95		8			13'—3½"		90000	10G
198	646						9.60	15				14'—7¼"		99000	11G
216	700						10.25			8		15'—11"		108000	12G
234	764		6		5		10.95					17'—2¾"		117000	13G
252	820						11.60		10			18'—6½"		126000	14G
270	880						12.25					19'—10¼"		135000	15G
288	940						12.90			10		21'—2"		144000	16G
306	1000						13.50					22'—5½"		153000	17G
324	1060						14.10					23'—9¼"		162000	18G
342	1120						14.70	20				25'—1"		171000	19G
360	1180		8		6		15.25		12			26'—4¾"		180000	20G
378	1230						15.80					27'—8½"		189000	21G
398	1300						16.20					29'—0"		199000	22G
416	1360						16.85					30'—3¾"		208000	23G

"Buffalo"

Sizes and Dimensions of Buffalo Standard Heaters

Number of Pipes	Length of Section	Section Number	Extreme Height of Section	Width of Section	Linear Feet of 1" Pipe per Section	Total Effective Square Feet of Heating Surface	Equivalent in Linear Feet of 1" Pipe per Section	Clear Area for Air Passage Sq. Ft.	Weight Pounds
56	3' 4 Row	1A	3'-4"	8½"	140	54.7	159	4.4	473
		2A	3'-10"		168	64.2	186	5.2	515
		3A	4'-4"		196	74.0	215	6.0	565
		4A	4'-10"		224	83.7	243	6.8	616
		5A	5'-4"		252	93.3	271	7.6	656
		6A	5'-10"		280	102.5	298	8.4	708
72	4' 4 Row	1B	5'-4"	8½"	320	119.0	346	9.7	819
		2B	5'-10"		356	131.5	382	10.7	877
		3B	6'-4"		392	143.9	418	11.2	938
		4B	6'-10"		428	156.5	455	12.6	1003
80	4'-6" 4 Row	1C	5'-10"	8½"	396	148.2	431	12.1	997
		2C	6'-4"		436	162.0	480	13.1	1055
		3C	6'-10"		476	174.8	507	14.2	1127
		4C	7'-4"		516	188.6	548	15.3	1174
88	5' 4 Row	1D	6'-4"	8½"	476	174.3	507	14.1	1182
		2D	6'-10"		520	189.3	550	15.4	1262
		3D	7'-4"		564	204.8	595	16.6	1325
		4D	7'-10"		608	219.8	638	17.7	1407
104	6' 4 Row	1E	7'-4"	8½"	674	245.0	712	19.8	1505
		2E	7'-10"		726	262.9	763	21.3	1600
		3E	8'-4"		778	280.8	816	22.7	1695
		4E	8'-10"		830	298.7	868	24.2	1770
128	7' 4 Row	1G	7'-4"	8½"	796	291.0	845	23.6	1845
		2G	7'-10"		860	313.2	910	25.4	1950
		3G	8'-4"		924	335.2	974	27.2	2055
		4G	8'-10"		988	357.2	1037	29.0	2160
		5G	9'-4"		1052	379.2	1101	30.7	2280
		6G	9'-10"		1116	401.2	1163	32.5	2380

All Buffalo Standard Heaters are regularly furnished in the return bend pattern. The open area pattern is furnished on special order only.

Note.—All heaters furnished in return bend pattern unless otherwise specified.

Friction of Air Through Buffalo Heaters

Air Measured at 70°F. and 29.92" Barometer.

Loss of Air Pressure in Inches of Water per Square Inch.

Velocity Through Clear Area	Number of Sections 1	2	3	4	5	6	7	8
300	0.009	0.017	0.026	0.035	0.043	0.052	0.060	0.069
400	0.015	0.031	0.046	0.062	0.077	0.092	0.108	0.123
500	0.024	0.049	0.073	0.095	0.104	0.144	0.168	0.192
600	0.035	0.069	0.104	0.138	0.173	0.207	0.242	0.276
700	0.047	0.094	0.141	0.188	0.235	0.282	0.329	0.376
800	0.061	0.123	0.184	0.245	0.306	0.368	0.429	0.490
900	0.078	0.155	0.233	0.311	0.388	0.466	0.544	0.621
1000	0.096	0.191	0.287	0.382	0.479	0.574	0.670	0.765
1100	0.116	0.232	0.347	0.463	0.579	0.695	0.810	0.926
1200	0.138	0.276	0.414	0.551	0.689	0.827	0.965	1.103
1300	0.162	0.324	0.486	0.648	0.810	0.972	1.133	1.296
1400	0.187	0.375	0.562	0.750	0.936	1.124	1.311	1.500
1500	0.215	0.431	0.646	0.861	1.077	1.293	1.508	1.722
1600	0.245	0.490	0.735	0.980	1.226	1.471	1.716	1.961
1700	0.277	0.555	0.831	1.110	1.387	1.664	1.940	2.218
1800	0.310	0.620	0.930	1.240	1.550	1.860	2.167	2.480

Final Temperatures and Condensations

Buffalo Standard Heater

0 LBS.

0 lbs. Steam Pressure **212.0° F**

Velocity of Air in Feet per Minute — Measured at 70° F. and 29.92" Barometer

Temperature of Air Entering °F	Number of Heater Sections	600 Final Temperature	600 Condensation per Lineal Foot per Hour, Pounds	800 F. T.	800 C.	1000 F. T.	1000 C.	1200 F. T.	1200 C.	1400 F. T.	1400 C.	1600 F. T.	1600 C.	1800 F. T.	1800 C.
20°	1	46	.487	44	.600	42	.687	40	.752	39	.831	38	.901	37	.958
	2	69	.459	65	.562	61	.640	59	.730	56	.786	54	.849	52	.900
	3	88	.426	83	.525	79	.614	75	.685	72	.756	68	.797	65	.842
	4	105	.400	99	.491	94	.572	90	.656	86	.723	82	.776	78	.816
	5	120	.375	113	.465	107	.544	102	.616	98	.684	94	.742	90	.789
	6	133	.351	125	.438	119	.515	113	.583	109	.648	104	.701	101	.761
	7	143	.330	136	.413	129	.486	124	.555	119	.618	114	.670	110	.725
	8	152	.310	145	.390	139	.461	133	.531	128	.590	123	.645	119	.696
30°	1	55	.469	52	.550	51	.656	49	.711	48	.788	47	.851	46	.902
	2	77	.440	72	.525	69	.609	66	.674	65	.765	62	.799	60	.844
	3	95	.406	90	.500	86	.583	82	.650	79	.715	75	.718	73	.805
	4	111	.380	105	.469	100	.540	96	.620	92	.680	88	.726	85	.775
	5	125	.357	118	.440	112	.513	108	.586	104	.650	99	.692	96	.744
	6	137	.334	130	.417	123	.484	118	.550	114	.613	109	.659	106	.711
	7	147	.313	140	.392	133	.459	128	.525	123	.581	119	.635	115	.684
	8	155	.293	149	.372	142	.438	136	.499	132	.558	127	.606	123	.654
40°	1	64	.450	61	.525	60	.625	58	.676	57	.745	56	.801	55	.845
	2	84	.412	80	.500	78	.593	74	.636	73	.721	70	.749	68	.787
	3	102	.387	96	.466	93	.551	89	.612	87	.685	83	.715	81	.767
	4	116	.356	110	.438	106	.510	102	.582	99	.646	95	.689	92	.732
	5	129	.334	123	.415	118	.488	111	.505	110	.615	105	.652	103	.710
	6	140	.312	134	.392	128	.458	123	.519	119	.576	115	.626	112	.676
	7	150	.291	144	.371	138	.437	132	.492	128	.550	124	.600	120	.641
	8	158	.276	152	.350	146	.414	141	.475	136	.525	132	.575	128	.620
50°	1	72	.412	70	.500	69	.594	67	.638	66	.700	65	.751	64	.789
	2	91	.384	88	.475	85	.546	82	.599	80	.655	78	.699	76	.730
	3	108	.362	104	.450	99	.510	96	.575	93	.627	91	.682	88	.711
	4	122	.338	116	.413	112	.479	108	.545	105	.603	102	.652	99	.690
	5	134	.316	128	.391	123	.457	119	.519	115	.570	112	.622	109	.665
	6	145	.297	139	.371	133	.432	129	.494	124	.540	121	.592	117	.640
	7	154	.278	148	.349	142	.410	137	.466	132	.513	129	.564	125	.604
	8	162	.262	156	.331	150	.391	145	.447	140	.492	136	.538	132	.576
60°	1	81	.389	79	.474	77	.531	76	.600	75	.656	74	.701	73	.732
	2	99	.363	95	.438	93	.515	91	.580	89	.631	87	.674	85	.702
	3	114	.339	110	.415	106	.479	103	.538	100	.585	98	.632	96	.674
	4	128	.316	122	.388	118	.449	114	.508	111	.559	109	.614	106	.648
	5	139	.297	133	.366	129	.432	124	.481	121	.534	118	.582	115	.620
	6	149	.278	143	.346	138	.406	133	.458	130	.510	127	.560	123	.592
	7	157	.260	151	.325	146	.384	141	.434	137	.482	135	.535	130	.564
	8	165	.245	159	.309	153	.364	149	.419	144	.462	141	.506	136	.535
70°	1	90	.375	88	.450	86	.500	85	.562	84	.613	83	.651	82	.676
	2	106	.337	103	.412	100	.468	98	.525	96	.569	95	.624	93	.646
	3	120	.312	116	.383	115	.447	110	.500	108	.554	105	.582	104	.620
	4	133	.295	128	.363	121	.417	120	.469	117	.515	115	.564	113	.605
	5	144	.278	139	.345	134	.400	130	.451	126	.491	124	.541	121	.575
	6	153	.259	148	.325	143	.380	138	.425	134	.466	132	.518	129	.555
	7	161	.243	155	.303	150	.356	146	.407	142	.450	139	.492	135	.523
	8	168	.230	162	.287	157	.340	153	.390	148	.426	144	.463	139	.485

Final Temperatures and Condensations

Buffalo Standard Heater

5 Lbs.

5 lbs. Steam Pressure 227.0° F

Temperature of Air Entering	Number of Heater Sections	600 Final Temperature	600 Condensation per Lineal Foot per Hour	800 F. T.	800 C.	1000 F. T.	1000 C.	1200 F. T.	1200 C.	1400 F. T.	1400 C.	1600 F. T.	1600 C.	1800 F. T.	1800 C.
°F			Pounds												
20°	1	48	.536	46	.652	44	.759	42	.845	40	.907	39	.961	38	1.023
	2	73	.499	68	.610	64	.695	62	.795	59	.862	57	.935	54	.956
	3	94	.467	88	.573	83	.661	79	.745	75	.812	72	.875	69	.926
	4	112	.434	105	.536	99	.625	94	.702	90	.774	87	.847	83	.895
	5	127	.407	120	.505	113	.589	108	.667	103	.737	100	.840	95	.855
	6	141	.381	133	.476	126	.558	121	.637	115	.702	111	.766	107	.823
	7	152	.358	144	.449	137	.526	131	.600	126	.668	121	.726	117	.785
	8	162	.337	155	.425	147	.502	141	.573	135	.635	131	.700	126	.752
30°	1	57	.512	55	.632	52	.728	51	.796	50	.885	48	.910	47	.967
	2	80	.475	76	.581	72	.664	69	.739	67	.819	65	.885	63	.937
	3	100	.441	94	.539	89	.620	86	.706	83	.780	80	.840	77	.889
	4	117	.412	110	.506	104	.587	100	.665	96	.730	93	.796	90	.853
	5	132	.387	124	.476	118	.556	113	.631	109	.700	105	.760	102	.820
	6	145	.363	137	.450	130	.526	125	.600	120	.664	116	.724	112	.776
	7	156	.340	147	.421	141	.500	135	.568	130	.630	126	.691	122	.745
	8	165	.319	157	.400	150	.473	144	.539	139	.600	135	.663	131	.716
40°	1	66	.494	64	.606	62	.696	60	.759	59	.840	57	.860	56	.910
	2	88	.455	83	.544	80	.631	77	.700	75	.775	73	.834	71	.880
	3	106	.416	100	.505	96	.589	92	.656	90	.735	87	.790	85	.851
	4	123	.393	115	.474	111	.560	106	.626	103	.696	100	.758	97	.810
	5	137	.368	129	.450	124	.532	119	.601	115	.661	111	.719	108	.775
	6	149	.342	141	.425	135	.500	130	.569	125	.626	121	.682	118	.738
	7	159	.321	151	.400	145	.473	139	.536	134	.593	131	.655	127	.705
	8	168	.303	160	.378	154	.450	148	.511	143	.568	140	.630	135	.672
50°	1	74	.455	72	.556	70	.633	69	.720	67	.752	66	.809	65	.854
	2	95	.427	90	.505	87	.585	85	.664	83	.730	81	.784	79	.824
	3	112	.391	107	.479	103	.556	100	.634	96	.676	94	.740	92	.795
	4	129	.374	121	.450	117	.529	113	.598	109	.652	106	.708	104	.768
	5	142	.349	134	.425	129	.500	124	.563	120	.620	117	.679	114	.729
	6	153	.325	145	.398	140	.473	134	.531	130	.590	127	.648	123	.690
	7	163	.305	155	.378	149	.445	144	.509	139	.561	136	.619	132	.665
	8	171	.286	164	.359	157	.421	152	.483	147	.534	144	.593	140	.638
60°	1	83	.434	81	.530	79	.600	78	.682	76	.710	75	.759	74	.796
	2	102	.401	99	.493	98	.568	93	.625	91	.685	89	.733	87	.767
	3	119	.373	114	.455	110	.535	107	.593	104	.647	101	.689	100	.755
	4	134	.349	128	.431	123	.496	119	.562	116	.618	113	.670	110	.710
	5	146	.326	140	.405	135	.473	130	.536	126	.587	124	.638	120	.684
	6	157	.306	151	.381	145	.448	140	.506	136	.559	132	.606	129	.653
	7	167	.288	160	.361	154	.425	149	.482	145	.534	140	.576	137	.624
	8	175	.272	168	.340	162	.403	157	.460	153	.512	148	.555	144	.595
70°	1	91	.398	89	.506	87	.558	86	.606	85	.665	84	.708	83	.740
	2	109	.370	105	.442	102	.505	101	.586	98	.625	97	.683	95	.710
	3	126	.353	121	.420	116	.484	113	.553	110	.590	109	.656	107	.700
	4	140	.331	133	.398	129	.465	125	.522	122	.575	119	.620	117	.668
	5	151	.307	144	.375	140	.444	135	.495	131	.540	129	.598	126	.639
	6	161	.287	154	.353	149	.415	144	.468	140	.515	137	.564	135	.615
	7	169	.267	163	.335	157	.392	152	.445	148	.492	145	.540	142	.584
	8	177	.253	171	.318	165	.374	160	.426	156	.474	152	.516	149	.560

Velocity of Air in Feet per Minute. Measured at 70° F. and 29.92" Barometer

"Buffalo"

Final Temperatures and Condensations

Buffalo Standard Heater

20 LBS.

20 lbs. Steam Pressure 258.8° F

Temperature of Air Entering	Number of Heater Sections	600 Final Temperature	600 Condensation per Lineal Foot per Hour	800 F. T.	800 C.	1000 F. T.	1000 C.	1200 F. T.	1200 C.	1400 F. T.	1400 C.	1600 F. T.	1600 C.	1800 F. T.	1800 C.
°F			Pounds												
20°	1	52	.620	50	.775	47	.873	45	.970	43	1.052	42	1.138	41	1.220
	2	80	.584	75	.710	71	.823	68	.930	65	1.002	62	1.085	59	1.135
	3	101	.542	97	.668	92	.774	87	.866	83	.948	80	1.031	76	1.084
	4	125	.506	117	.630	110	.726	105	.821	100	.904	96	.983	91	1.030
	5	143	.478	134	.591	127	.694	120	.778	115	.864	110	.932	105	.990
	6	158	.445	149	.556	141	.652	134	.736	128	.816	124	.896	118	.948
	7	171	.418	162	.522	154	.617	147	.701	141	.781	136	.851	130	.921
	8	183	.394	174	.496	166	.590	158	.660	152	.747	146	.815	141	.881
30°	1	60	.580	58	.724	56	.840	54	.933	53	1.010	51	1.081	50	1.162
	2	87	.552	82	.671	78	.776	75	.873	72	.949	70	1.032	67	1.078
	3	111	.522	104	.646	99	.742	94	.825	91	.918	87	.982	84	1.026
	4	131	.488	123	.605	116	.694	110	.775	107	.870	102	.930	98	.987
	5	147	.455	139	.566	132	.661	125	.740	121	.827	116	.891	112	.955
	6	162	.426	153	.530	146	.625	138	.697	134	.785	129	.853	124	.910
	7	174	.398	166	.500	158	.589	151	.668	145	.742	140	.810	136	.879
	8	186	.378	177	.475	169	.561	162	.640	156	.713	151	.782	145	.838
40°	1	69	.561	66	.672	64	.775	63	.894	62	.995	60	1.034	59	1.105
	2	95	.534	90	.645	86	.744	83	.834	81	.926	78	.981	76	1.048
	3	117	.496	111	.610	105	.699	100	.775	98	.873	95	.947	91	.987
	4	136	.464	129	.579	122	.662	117	.746	113	.825	109	.892	105	.943
	5	152	.435	144	.540	137	.630	131	.708	127	.791	122	.850	118	.909
	6	166	.406	158	.508	150	.593	144	.672	139	.746	134	.810	130	.870
	7	178	.381	169	.475	162	.561	155	.635	150	.710	145	.773	141	.837
	8	188	.358	180	.452	172	.534	166	.610	160	.680	155	.743	150	.801
50°	1	78	.542	75	.646	73	.743	72	.855	70	.935	69	.982	68	1.016
	2	102	.504	98	.620	94	.711	91	.795	89	.881	87	.955	84	.990
	3	124	.477	118	.585	112	.667	108	.749	105	.828	102	.896	99	.948
	4	142	.444	135	.553	129	.638	123	.708	119	.780	116	.854	112	.900
	5	157	.415	149	.514	142	.597	137	.677	132	.745	129	.818	125	.874
	6	170	.387	162	.482	155	.565	149	.640	144	.709	140	.776	136	.832
	7	182	.365	174	.456	166	.534	160	.607	154	.671	150	.736	146	.795
	8	192	.344	184	.432	176	.500	170	.581	164	.645	159	.705	155	.765
60°	1	87	.522	84	.620	82	.711	80	.778	79	.860	78	.930	77	.988
	2	110	.485	105	.582	102	.678	99	.756	96	.815	95	.894	93	.960
	3	129	.445	124	.551	119	.635	115	.710	111	.767	109	.844	106	.890
	4	147	.425	140	.520	134	.597	130	.679	125	.734	122	.801	119	.856
	5	162	.398	155	.493	148	.570	143	.645	138	.710	135	.776	131	.826
	6	175	.371	167	.460	160	.538	154	.607	149	.671	145	.733	142	.793
	7	186	.348	178	.435	171	.511	165	.580	159	.637	155	.699	151	.754
	8	195	.327	187	.410	180	.485	174	.553	168	.611	164	.672	159	.720
70°	1	95	.485	93	.595	91	.679	89	.730	88	.814	87	.878	86	.930
	2	117	.456	113	.555	110	.646	106	.698	105	.791	102	.827	101	.903
	3	136	.425	131	.525	126	.603	122	.671	119	.737	116	.793	114	.852
	4	152	.396	146	.495	141	.573	135	.630	132	.700	129	.763	125	.813
	5	166	.373	159	.462	153	.538	148	.607	144	.673	140	.725	137	.780
	6	178	.348	171	.435	165	.512	159	.575	154	.634	151	.698	147	.745
	7	189	.329	181	.408	175	.483	169	.545	164	.605	160	.662	156	.713
	8	198	.314	191	.391	184	.460	178	.523	172	.577	168	.634	164	.684

Velocity of Air in Feet per Minute. Measured at 70° F. and 29.92" Barometer

"Buffalo"

Final Temperatures and Condensations

Buffalo Standard Heater

40 LBS.

40 lbs. Steam Pressure 286.7° F

Velocity of Air in Feet per Minute — Measured at 70° and 29.92" Barometer

Temperature of Air Entering	Number of Heater Sections	600 Final Temperature	600 Condensation per Lineal Foot per Hour	800 F. T.	800 C.	1000 F. T.	1000 C.	1200 F. T.	1200 C.	1400 F. T.	1400 C.	1600 F. T.	1600 C.	1800 F. T.	1800 C.
°F			Pounds												
20°	1	55	.695	52	.845	50	.995	47	1.073	46	1.205	44	1.272	43	1.370
	2	87	.655	81	.806	76	.926	72	1.032	69	1.138	66	1.218	63	1.282
	3	113	.615	106	.758	99	.874	94	.980	89	1.065	86	1.168	83	1.250
	4	136	.574	127	.707	120	.827	113	.924	108	1.018	104	1.110	99	1.179
	5	155	.537	146	.668	138	.784	131	.884	125	.975	120	1.060	115	1.133
	6	173	.505	163	.630	154	.735	146	.831	139	.916	134	1.005	129	1.080
	7	188	.475	177	.592	168	.699	160	.792	153	.877	148	.967	142	1.037
	8	200	.446	109	.562	181	.666	172	.756	165	.839	160	.927	153	.993
30°	1	64	.675	61	.819	59	.961	56	1.032	55	1.160	53	1.220	52	1.310
	2	93	.625	88	.767	84	.894	80	.991	76	1.068	74	1.165	72	1.252
	3	120	.595	112	.724	106	.840	102	.954	97	1.035	93	1.113	90	1.190
	4	141	.550	133	.681	126	.794	120	.894	115	.984	110	1.058	106	1.132
	5	160	.517	151	.642	143	.750	137	.851	132	.947	126	1.019	121	1.085
	6	177	.485	168	.608	159	.708	152	.805	146	.893	140	.968	135	1.040
	7	191	.456	182	.574	173	.676	165	.765	158	.843	152	.922	147	.994
	8	203	.429	194	.542	185	.642	176	.727	169	.804	164	.888	158	.955
40°	1	73	.655	70	.793	67	.895	65	.994	64	1.111	62	1.167	61	1.251
	2	101	.605	96	.740	91	.834	88	.952	85	1.043	82	1.111	80	1.192
	3	126	.569	119	.697	113	.806	109	.915	105	1.003	101	1.078	97	1.131
	4	147	.530	139	.654	132	.760	127	.854	122	.949	117	1.019	113	1.088
	5	165	.498	156	.615	149	.724	143	.820	138	.909	132	.976	127	1.039
	6	181	.465	172	.581	164	.680	157	.772	151	.855	145	.924	140	.990
	7	195	.439	186	.551	177	.648	169	.731	163	.811	157	.885	152	.952
	8	207	.415	197	.519	189	.617	180	.697	174	.775	168	.847	162	.940
50°	1	82	.635	79	.766	76	.861	74	.955	73	1.065	71	1.113	70	1.191
	2	108	.575	104	.711	100	.826	96	.913	93	.998	91	1.085	89	1.163
	3	133	.549	126	.671	120	.774	116	.875	112	.957	108	1.026	105	1.091
	4	153	.510	145	.628	138	.726	133	.824	129	.915	124	.979	120	1.042
	5	170	.478	162	.594	154	.690	148	.780	143	.864	138	.934	134	1.000
	6	185	.445	177	.559	168	.648	162	.739	156	.815	151	.889	146	.950
	7	198	.419	189	.525	181	.619	173	.697	167	.771	162	.846	157	.909
	8	210	.397	201	.500	192	.588	184	.667	178	.740	172	.807	167	.873
60°	1	90	.596	88	.740	85	.829	83	.916	82	1.019	80	1.060	79	1.131
	2	116	.556	111	.671	108	.795	104	.873	101	.950	99	1.032	96	1.072
	3	138	.516	132	.625	127	.740	122	.821	118	.896	116	.990	112	1.031
	4	157	.480	150	.596	144	.693	138	.774	134	.856	130	.925	127	.997
	5	175	.458	167	.568	159	.656	153	.740	148	.816	144	.891	140	.954
	6	189	.425	180	.530	173	.620	166	.699	160	.769	156	.845	151	.902
	7	202	.402	193	.506	186	.595	178	.669	172	.739	167	.809	161	.858
	8	213	.380	204	.476	196	.564	189	.642	181	.700	177	.775	171	.828
70°	1	98	.556	96	.686	94	.795	92	.875	91	.975	89	1.008	88	1.074
	2	124	.536	119	.648	115	.745	113	.853	110	.927	107	.980	105	1.042
	3	145	.496	139	.609	134	.707	130	.795	127	.880	123	.937	120	.992
	4	163	.460	157	.575	150	.661	146	.751	142	.834	137	.886	133	.939
	5	180	.448	172	.541	165	.630	159	.709	155	.789	150	.849	146	.906
	6	194	.400	186	.511	178	.592	171	.666	166	.747	164	.800	157	.861
	7	206	.385	197	.480	190	.567	182	.635	177	.705	172	.771	167	.824
	8	216	.365	208	.456	201	.542	193	.613	187	.676	182	.741	177	.798

Final Temperatures and Condensations

Buffalo Standard Heater

60 LBS.

60 lbs. Steam Pressure 307.3° F.

Velocity of Air in Feet per Minute — Measured at 70° F. and 29.92" Barometer

Temperature of Air Entering	Number of Heater Sections	600 Final Temperature	600 Condensation per Lineal Foot per Hour	800 F. T.	800 C.	1000 F. T.	1000 C.	1200 F. T.	1200 C.	1400 F. T.	1400 C.	1600 F. T.	1600 C.	1800 F. T.	1800 C.
°F.			Pounds												
20°	1	58	.761	55	.934	52	1.070	50	1.201	48	1.310	46	1.390	44	1.440
	2	91	.715	85	.874	80	1.007	76	1.129	72	1.225	69	1.318	66	1.390
	3	119	.670	111	.819	105	.956	99	1.075	94	1.168	90	1.262	86	1.339
	4	141	.625	134	.768	127	.900	120	1.109	114	1.107	109	1.197	104	1.270
	5	166	.586	155	.724	146	.843	139	.958	132	1.050	126	1.138	121	1.218
	6	184	.551	173	.684	164	.806	155	.909	148	1.005	141	1.085	136	1.160
	7	200	.516	188	.644	179	.764	170	.861	163	.959	155	1.035	150	1.120
	8	214	.489	202	.611	192	.722	183	.822	176	.919	168	.996	162	1.073
30°	1	67	.744	63	.907	61	1.035	59	1.161	57	1.262	55	1.336	53	1.380
	2	98	.686	92	.834	88	.975	84	1.088	81	1.200	78	1.290	75	1.360
	3	126	.650	118	.792	111	.911	107	1.049	102	1.136	98	1.225	93	1.279
	4	150	.606	141	.747	132	.858	126	.969	120	1.060	115	1.143	111	1.225
	5	170	.562	160	.696	151	.810	145	.925	138	1.013	132	1.094	127	1.170
	6	187	.528	177	.656	167	.767	161	.881	154	.975	147	1.049	142	1.130
	7	203	.496	192	.620	182	.729	175	.833	167	.919	160	.996	155	1.079
	8	215	.466	205	.588	195	.693	187	.792	180	.883	173	.962	167	1.036
40°	1	76	.721	72	.854	70	1.002	68	1.122	66	1.218	64	1.283	62	1.320
	2	106	.656	100	.806	95	.925	92	1.049	89	1.153	86	1.238	83	1.300
	3	132	.622	125	.765	118	.878	114	1.009	109	1.089	105	1.171	101	1.238
	4	153	.570	147	.697	138	.825	133	.959	127	1.025	122	1.103	118	1.180
	5	174	.538	165	.670	157	.783	150	.885	144	.975	138	1.051	134	1.134
	6	192	.511	182	.634	173	.746	166	.847	159	.934	153	1.013	148	1.090
	7	206	.476	196	.597	187	.705	179	.799	172	.885	166	.966	160	1.045
	8	219	.451	209	.568	199	.668	192	.766	184	.848	178	.929	172	.997
50°	1	83	.681	81	.826	78	.935	76	1.042	75	1.170	73	1.230	71	1.261
	2	113	.635	108	.780	103	.891	99	.987	97	1.107	94	1.182	91	1.240
	3	138	.595	132	.738	125	.844	120	.953	116	1.040	112	1.119	109	1.196
	4	160	.555	152	.665	144	.791	139	.899	133	.977	129	1.062	125	1.134
	5	179	.518	170	.643	162	.750	156	.853	150	.937	144	1.009	140	1.086
	6	195	.487	186	.607	177	.711	170	.807	164	.895	158	.967	153	1.039
	7	209	.456	200	.574	191	.675	183	.764	177	.851	171	.927	166	1.000
	8	222	.433	212	.545	203	.643	195	.731	188	.813	182	.889	177	.960
60°	1	93	.664	90	.800	87	.902	85	1.003	83	1.075	82	1.175	80	1.201
	2	121	.615	115	.740	111	.857	107	.947	104	1.035	102	1.130	99	1.179
	3	145	.575	138	.702	132	.810	127	.912	123	.994	119	1.064	116	1.135
	4	166	.535	158	.659	151	.765	145	.858	140	.942	135	1.010	132	1.089
	5	185	.502	176	.622	168	.723	161	.813	155	.890	150	.965	146	1.039
	6	201	.471	191	.585	183	.689	175	.773	169	.856	163	.923	159	.998
	7	214	.442	205	.554	196	.652	188	.735	182	.817	175	.882	171	.958
	8	226	.418	216	.524	208	.622	200	.706	193	.783	187	.855	181	.915
70°	1	101	.621	99	.771	96	.868	94	.963	92	1.030	91	1.121	89	1.141
	2	128	.585	124	.725	119	.824	116	.926	113	1.013	110	1.075	107	1.119
	3	151	.548	145	.675	139	.776	135	.885	130	.946	127	1.028	123	1.075
	4	172	.515	164	.632	157	.731	152	.848	147	.906	142	.969	139	1.043
	5	189	.478	181	.595	173	.690	167	.780	161	.852	156	.924	152	.989
	6	204	.450	195	.558	187	.655	181	.746	174	.816	169	.887	165	.958
	7	217	.421	208	.528	200	.624	193	.706	186	.777	181	.851	176	.906
	8	229	.400	219	.591	212	.596	204	.676	197	.748	191	.814	186	.876

Final Temperatures and Condensations

Buffalo Standard Heater

80 LBS.

80 lbs. Steam Pressure 323.7° F

Velocity of Air in Feet per Minute — Measured at 70° F. and 29.92" Barometer

Temperature of Air Entering	Number of Heater Sections	600 Final Temperature	600 Condensation per Lineal Foot per Hour	800 F. T.	800 C.	1000 F. T.	1000 C.	1200 F. T.	1200 C.	1400 F. T.	1400 C.	1600 F. T.	1600 C.	1800 F. T.	1800 C.
°F			Pounds												
20°	1	60	.815	57	1.006	54	1.155	52	1.306	49	1.380	48	1.522	46	1.591
	2	95	.769	88	.925	83	1.072	79	1.205	75	1.310	72	1.419	69	1.502
	3	124	.709	116	.871	109	1.039	103	1.130	98	1.240	94	1.343	90	1.430
	4	151	.670	141	.825	132	.956	125	1.072	119	1.182	113	1.269	109	1.357
	5	173	.624	162	.770	153	.903	145	1.019	138	1.121	132	1.216	126	1.294
	6	193	.589	181	.730	171	.858	162	.966	155	1.071	148	1.160	142	1.245
	7	210	.554	198	.691	187	.813	178	.924	170	1.021	163	1.112	156	1.190
	8	224	.520	212	.653	201	.769	192	.879	185	.984	177	1.069	169	1.139
30°	1	69	.795	65	.952	62	1.088	60	1.223	58	1.332	57	1.460	55	1.530
	2	103	.748	96	.899	91	1.039	86	1.145	83	1.264	80	1.363	77	1.440
	3	132	.695	123	.844	116	.975	110	1.089	105	1.191	101	1.289	97	1.369
	4	157	.643	146	.790	138	.922	131	1.032	125	1.136	120	1.228	115	1.305
	5	179	.606	167	.744	158	.868	150	.979	144	1.081	138	1.172	132	1.245
	6	196	.565	185	.704	176	.830	167	.942	160	1.031	154	1.124	147	1.192
	7	213	.534	201	.665	191	.783	182	.889	175	.990	168	1.073	161	1.147
	8	227	.502	215	.639	205	.744	195	.843	188	.942	181	1.028	174	1.100
40°	1	77	.755	74	.925	71	1.052	69	1.182	67	1.286	66	1.414	64	1.470
	2	109	.707	104	.874	99	1.004	94	1.104	91	1.245	88	1.310	86	1.410
	3	138	.667	130	.816	123	.941	117	1.049	112	1.144	108	1.234	105	1.329
	4	162	.618	153	.770	145	.896	137	.992	132	1.100	126	1.172	123	1.275
	5	182	.578	173	.721	164	.841	155	.938	150	1.043	144	1.129	139	1.206
	6	200	.544	190	.680	181	.801	172	.897	166	1.000	159	1.079	153	1.153
	7	216	.513	205	.641	195	.754	187	.860	179	.947	173	1.035	167	1.111
	8	230	.484	219	.608	209	.719	200	.818	192	.905	185	.986	179	1.062
50°	1	86	.734	83	.870	80	1.020	78	1.141	76	1.239	75	1.360	73	1.409
	2	117	.686	111	.840	107	.970	102	1.062	99	1.168	96	1.255	94	1.350
	3	144	.640	136	.781	130	.907	124	1.008	119	1.096	115	1.180	112	1.267
	4	167	.593	159	.743	151	.862	144	.961	138	1.051	133	1.132	129	1.212
	5	187	.558	178	.695	169	.807	161	.905	155	.996	150	1.085	145	1.159
	6	204	.524	194	.653	185	.766	177	.864	171	.960	165	1.042	159	1.111
	7	219	.493	209	.618	200	.730	191	.825	184	.914	178	.996	172	1.069
	8	233	.466	222	.585	212	.689	204	.787	196	.870	190	.952	184	1.024
60°	1	95	.714	91	.844	89	.986	87	1.105	85	1.190	84	1.305	82	1.347
	2	124	.655	119	.803	114	.920	111	1.041	107	1.120	104	1.200	102	1.288
	3	150	.612	143	.753	137	.873	131	.966	127	1.065	123	1.143	120	1.225
	4	173	.572	164	.708	157	.827	150	.920	145	1.016	140	1.091	136	1.168
	5	192	.537	183	.667	174	.774	167	.873	161	.958	156	1.043	151	1.110
	6	209	.507	199	.631	190	.739	182	.830	176	.920	170	.996	165	1.071
	7	224	.479	213	.595	204	.701	196	.795	189	.879	183	.957	177	1.024
	8	237	.451	226	.565	217	.668	208	.756	201	.840	195	.918	189	.986
70°	1	103	.673	100	.815	98	.953	96	1.064	94	1.144	92	1.251	91	1.286
	2	132	.635	127	.775	122	.886	118	.980	115	1.072	111	1.119	110	1.228
	3	157	.591	150	.726	144	.840	139	.939	134	1.018	130	1.090	127	1.165
	4	178	.548	170	.682	163	.794	157	.890	151	.969	147	1.050	143	1.127
	5	197	.517	188	.640	180	.746	173	.840	167	.926	162	1.000	157	1.060
	6	213	.486	204	.608	195	.710	188	.802	181	.884	175	.950	170	1.020
	7	227	.459	217	.574	208	.672	200	.760	193	.838	188	.919	182	.981
	8	240	.434	229	.540	220	.638	212	.725	205	.805	199	.878	195	.940

"Buffalo"

Final Temperatures and Condensations

Buffalo Standard Heater

100 LBS.

100 lbs. Steam Pressure 337.6° F

Velocity of Air in Feet per Minute — Measured at 70° F. and 29.92" Barometer

Temperature of Air Entering	Number of Heater Sections	600 Final Temperature	600 Condensation per Lineal Foot per Hour	800 F. T.	800 C.	1000 F. T.	1000 C.	1200 F. T.	1200 C.	1400 F. T.	1400 C.	1600 F. T.	1600 C.	1800 F. T.	1800 C.
°F			Pounds												
20°	1	62	.869	58	1.049	55	1.209	52	1.319	50	1.449	48	1.543	47	1.680
	2	98	.805	91	.979	86	1.139	81	1.260	77	1.371	74	1.486	71	1.578
	3	129	.752	120	.920	112	1.060	106	1.190	101	1.303	96	1.400	92	1.491
	4	156	.704	145	.861	137	1.009	129	1.129	123	1.242	117	1.335	112	1.427
	5	179	.655	168	.814	158	.947	150	1.072	143	1.184	136	1.274	130	1.360
	6	200	.620	188	.773	177	.903	168	1.016	160	1.128	153	1.223	147	1.315
	7	218	.584	205	.728	194	.855	185	.976	176	1.076	169	1.172	162	1.285
	8	233	.554	220	.690	209	.815	199	.928	190	1.027	183	1.125	175	1.202
30°	1	71	.847	67	1.020	63	1.139	61	1.279	59	1.400	57	1.489	55	1.554
	2	106	.784	98	.937	93	1.086	88	1.199	84	1.300	82	1.430	78	1.485
	3	136	.731	126	.883	119	1.028	113	1.148	107	1.240	103	1.345	99	1.390
	4	162	.683	152	.840	142	.967	135	1.085	128	1.182	124	1.294	118	1.363
	5	183	.638	173	.786	164	.920	156	1.040	148	1.136	142	1.231	136	1.310
	6	201	.600	192	.745	182	.874	173	.989	165	1.088	158	1.177	152	1.263
	7	221	.564	208	.700	198	.826	189	.940	181	1.042	173	1.126	167	1.212
	8	236	.533	223	.665	212	.785	203	.897	194	.991	187	1.082	180	1.163
40°	1	80	.827	76	.994	72	1.102	70	1.234	68	1.351	66	1.434	64	1.494
	2	113	.754	106	.910	100	1.034	96	1.157	92	1.252	90	1.375	87	1.455
	3	142	.704	134	.865	126	.995	120	1.105	115	1.200	110	1.290	107	1.389
	4	167	.657	158	.813	148	.932	141	1.044	135	1.148	130	1.240	125	1.319
	5	189	.614	178	.759	169	.886	161	1.000	154	1.098	148	1.188	142	1.263
	6	208	.579	196	.718	187	.845	178	.954	170	1.046	164	1.140	158	1.221
	7	224	.542	212	.676	202	.796	194	.912	186	1.008	179	1.094	172	1.169
	8	239	.515	226	.642	216	.760	207	.865	199	.960	192	1.049	185	1.125
50°	1	89	.806	84	.939	81	1.070	79	1.194	77	1.305	75	1.379	73	1.431
	2	120	.723	113	.868	108	1.000	104	1.115	100	1.204	97	1.293	95	1.391
	3	148	.676	140	.828	133	.959	127	1.053	122	1.160	118	1.253	114	1.326
	4	173	.636	163	.779	155	.906	148	1.013	142	1.110	137	1.199	132	1.271
	5	194	.593	183	.731	174	.852	167	.965	160	1.060	154	1.142	149	1.225
	6	213	.562	201	.694	191	.810	184	.926	176	1.015	170	1.103	164	1.180
	7	228	.525	216	.653	206	.767	198	.875	190	.965	183	1.048	177	1.123
	8	242	.497	229	.618	220	.734	214	.835	203	.925	196	1.007	190	1.086
60°	1	97	.765	93	.910	90	1.035	88	1.154	86	1.255	84	1.323	82	1.369
	2	128	.702	122	.854	117	.981	113	1.075	109	1.180	105	1.230	103	1.330
	3	155	.655	147	.800	140	.924	135	1.036	130	1.127	125	1.196	122	1.285
	4	179	.615	169	.751	161	.871	154	.973	149	1.075	144	1.158	139	1.221
	5	199	.572	189	.709	180	.826	173	.932	166	1.021	160	1.100	155	1.180
	6	217	.541	206	.672	196	.782	189	.891	181	.974	175	1.058	170	1.139
	7	232	.507	221	.631	211	.743	203	.846	195	.931	188	1.007	182	1.080
	8	246	.481	234	.600	224	.707	216	.809	208	.895	201	.975	194	1.039
70°	1	105	.724	101	.855	99	1.000	96	1.071	94	1.159	93	1.269	91	1.306
	2	135	.671	129	.813	124	.930	120	1.033	116	1.100	111	1.210	111	[illegible]
	3	161	.628	154	.772	147	.893	142	.992	136	1.062	133	1.160	129	1.221
	4	184	.590	175	.724	168	.845	161	.941	155	1.026	150	1.102	146	1.179
	5	204	.551	193	.676	185	.790	178	.891	172	.982	166	1.055	161	1.130
	6	221	.520	210	.641	201	.753	193	.850	187	.942	181	1.021	175	1.087
	7	236	.486	224	.610	215	.713	207	.810	200	.897	193	.969	187	1.035
	8	249	.463	237	.576	228	.682	219	.773	212	.850	205	.934	199	1.000

"Buffalo"

Final Temperatures and Condensations

Vento Cast Iron Heater

Regular Section Standard Spacing **5" Centers of Sections** **Steam 5 lb. Gauge 227°**

Temperature of Entering Air	Number of Stacks Deep	Velocity Through Heater in Feet per Minute. Measured at 70° F. — 600: Final Temperature Air Leaving Heater	600: Cond. lbs. per Sq. Ft. per Hour	800: F. T.	800: C.	1000: F. T.	1000: C.	1200: F. T.	1200: C.	1400: F. T.	1400: C.	1600: F. T.	1600: C.	1800: F. T.	1800: C.
20°	1	58	1.46	54	1.75	51	1.99	49	2.23	47	2.42	45	2.56	43	2.65
	2	87	1.29	81	1.57	76	1.80	72	2.00	69	2.24	66	2.37	64	2.51
	3	110	1.15	103	1.42	97	1.65	92	1.85	88	2.06	83	2.22	82	2.38
	4	130	1.06	122	1.31	115	1.52	110	1.73	105	1.91	101	2.08	97	2.22
	5	144	.95	136	1.19	130	1.41	124	1.60	119	1.78	114	1.93	110	2.08
	6	156	.87	148	1.10	142	1.30	136	1.49	130	1.65	126	1.81	122	1.96
	7	167	.81	159	1.02	152	1.21	146	1.39	141	1.55	136	1.70	132	1.85
	8	175	.75	167	.94	161	1.13	155	1.30	150	1.46	145	1.60	141	1.74
30°	1	66	1.39	62	1.64	60	1.92	58	2.17	56	2.33	54	2.46	52	2.54
	2	93	1.21	87	1.46	83	1.70	79	1.89	76	2.06	73	2.21	71	2.37
	3	115	1.09	108	1.33	103	1.56	98	1.75	94	1.91	91	2.08	88	2.23
	4	134	1.00	126	1.23	120	1.44	115	1.63	110	1.80	106	1.95	102	2.08
	5	148	.91	140	1.13	134	1.33	128	1.51	123	1.67	118	1.80	115	1.96
	6	159	.83	151	1.04	145	1.23	139	1.40	134	1.56	130	1.71	126	1.85
	7	169	.76	161	.96	155	1.15	149	1.31	144	1.46	139	1.60	135	1.73
	8	177	.71	169	.89	163	1.07	158	1.23	153	1.38	148	1.51	144	1.64
40°	1	74	1.31	70	1.54	68	1.80	66	2.00	64	2.16	62	2.26	61	2.42
	2	100	1.15	94	1.39	90	1.60	86	1.77	83	1.93	81	2.10	79	2.27
	3	121	1.04	114	1.26	109	1.47	104	1.64	100	1.79	97	1.95	94	2.08
	4	138	.94	130	1.15	124	1.35	119	1.52	115	1.68	111	1.82	108	1.96
	5	151	.85	144	1.07	138	1.26	132	1.42	127	1.56	123	1.70	120	1.85
	6	162	.78	154	.97	148	1.15	143	1.32	138	1.47	134	1.60	131	1.75
	7	171	.72	164	.91	158	1.08	153	1.24	148	1.39	143	1.51	139	1.63
	8	179	.67	171	.84	165	1.00	160	1.15	155	1.29	151	1.42	147	1.54
60°	1	90	1.15	86	1.34	84	1.54	82	1.69	81	1.89	80	2.05	79	2.19
	2	112	1.00	107	1.21	103	1.38	100	1.54	98	1.71	96	1.85	94	1.96
	3	131	.91	124	1.09	120	1.28	116	1.44	113	1.58	110	1.71	108	1.85
	4	146	.83	139	1.01	134	1.19	129	1.33	125	1.46	122	1.59	119	1.70
	5	158	.75	151	.93	145	1.09	140	1.23	136	1.36	133	1.50	130	1.62
	6	167	.69	160	.85	155	1.02	150	1.15	146	1.29	142	1.40	139	1.52

Friction of Air Through Vento Cast Iron Heaters

Friction Loss in Inches of Water. **Air Measured at 70° F.**

Regular Section

Velocity Feet per Minute	Spacing of Sections Inches	Number of Stacks: 1	2	3	4	5	6	7	8
600	5	0.021	0.040	0.058	0.076	0.094	0.112	0.130	0.149
700	5	0.028	0.054	0.079	0.105	0.130	0.155	0.180	0.205
800	5	0.037	0.070	0.103	0.135	0.167	0.200	0.232	0.265
900	5	0.047	0.088	0.129	0.170	0.211	0.252	0.293	0.335
1000	5	0.059	0.109	0.160	0.211	0.262	0.313	0.364	0.415
1100	5	0.071	0.132	0.193	0.255	0.316	0.377	0.438	0.501
1200	5	0.084	0.157	0.230	0.303	0.376	0.449	0.522	0.596
1300	5	0.099	0.185	0.271	0.356	0.442	0.528	0.614	0.701
1400	5	0.115	0.214	0.314	0.414	0.513	0.612	0.712	0.813
1500	5	0.132	0.246	0.360	0.474	0.588	0.702	0.816	0.932
1600	5	0.150	0.280	0.410	0.540	0.670	0.809	0.930	1.060
1700	5	0.169	0.316	0.463	0.609	0.756	0.903	1.049	1.197
1800	5	0.190	0.354	0.518	0.683	0.848	1.012	1.177	1.342

Buffalo

Buffalo Single Vertical Engines—Class "A"

Maximum Horsepower Allowable for Corresponding Frame

High Pressure

Maximum Horsepower	Maximum R. P. M.	Cylinder Diameter and Stroke	Floor Space: Length	Floor Space: Width	Floor Space: Height	Standard Fly Wheel: Diameter	Standard Fly Wheel: Face	Steam and Exhaust Pipes: Steam	Steam and Exhaust Pipes: Exhaust	Shipping Weight
6	550	4 x 4	34	32	46	27	5½	1¼	1½	1260
12	475	5 x 5	37	34	55	31	6	1½	2	1740
20	450	6 x 6	41	37	65	33	6½	2	2½	2400
20	425	7 x 7	41	37	65	33	6½	2	2½	2800
45	400	8 x 8	43	40	78	39	7	2½	3	3270
45	400	10 x 8	43	40	78	39	7	3	3½	3420
65	350	8 x 10	52	52	96	49	11½	2½	3	6070
65	350	10 x 10	52	52	96	49	11½	3	3½	6240
65	350	12 x 10	52	52	96	49	11½	3½	4	6460
95	300	10 x 12	62	64	118	57	13	3½	4	8830
95	300	12 x 12	62	64	118	57	13	4	5	9000

Low Pressure

18	450	8 x 6	41	37	65	33	6½	2½	3	2450
45	400	12 x 8	43	40	78	39	7	3	3½	3780
45	400	13 x 8	43	40	78	39	7	3	3½	4160
45	400	15 x 8	43	40	78	39	7	3½	4	6490
65	350	15 x 10	52	52	96	49	11½	3	3½	7150
95	300	15 x 12	62	64	118	57	13	4	5	10830
95	300	18 x 12	62	64	118	57	13	5	6	11270

Buffalo Horizontal Engines, Center Crank—Class "A"

30	300	5 x 10	70	30	30	40	8½	1½	2½	1980
30	300	6 x 10	70	30	30	40	8½	1½	2½	2030
50	250	7 x 12	86	34	32	40	8½	2	3	2750
50	250	8 x 12	86	34	32	40	8½	2	3	2970
65	225	8 x 14	102	40	37	49	10	2½	3½	3850
65	225	9 x 14	102	40	37	49	10	2½	3½	4070

Buffalo Single Vertical Engines—Class "I"

Cylinder Below Shaft

Maximum Horsepower	Maximum R. P. M.	Cylinder Diameter and Stroke	Steam and Exhaust Pipes: Steam	Steam and Exhaust Pipes: Exhaust	Weight
5	600	3 x 3½	1	1¼	340
7½	500	4 x 3½	1	1¼	370
11	400	4½ x 5	1¼	1½	780
18½	325	5½ x 7	1¼	1½	1100
25	275	6½ x 8	1½	2	1500
30	220	7½ x 9	2	2½	2000

Buffalo

Planoidal Fans

With Proper Combinations of Heaters and Engines for Public Buildings and Industrial Installations

Fan Number	Cubic Feet of Air per Minute: 1" Static Pressure	Cubic Feet of Air per Minute: 2" Static Pressure	Buffalo Standard Heater: Arrangement	Buffalo Standard Heater: Size	Buffalo Standard Heater: Clear Area Square Feet	Engine: Low Pressure	Engine: High Pressure
50	4,550	6,440	Single	3'0"x3'4" 3'0"x3'10" 3'0"x4'4"	4.4 5.2 6.0	8 x 6	4x4A 4x3½I
55	5,500	7,780	Single	3'0"x3'10" 3'0"x4'4" 3'0"x4'10" 3'0"x5'4"	5.2 6.0 6.8 7.6	8 x 6	4x4A 4x3½I
60	6,550	9,260	Single	3'0"x4'10" 3'0"x5'4" 3'0"x5'10"	6.8 7.6 8.4	8 x 6	5x5A 4½x5I
70	8,930	12,630	Single	3'0"x5'10" 4'0"x5'4" 4'0"x5'10" 4'0"x6'4" 4'0"x6'10" 4'6"x5'10"	8.4 9.7 10.7 11.2 12.6 12.1	10 x 8	5x5A 4½x5I
80	11,630	16,450	Single	4'0"x6'4" 4'0"x6'10" 4'6"x5'10" 4'6"x6'4" 4'6"x6'10" 4'6"x7'4" 5'0"x6'4" 5'0"x6'10"	11.2 12.6 12.1 13.1 14.2 15.3 14.1 15.4	10 x 8	6x6A 5½x7I 5x10N
90	14,730	20,850	Single	4'6"x6'10" 4'6"x7'4" 5'0"x6'4" 5'0"x6'10" 5'0"x7'4" 5'0"x7'10" 6'0"x7'4"	14.2 15.3 14.1 15.4 16.6 17.7 19.8	10 x 8	7x7A 5½x7I 5x10N
100	18,200	25,750	Single	5'0"x7'4" 5'0"x7'10" 6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10"	16.6 17.7 19.8 21.3 22.7 24.2 23.6 25.4	12 x 8	7x7A 6½x8I 5x10N 6x10N
110	22,000	31,100	Single	6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10" 7'0"x8'4" 7'0"x8'10" 7'0"x9'4"	21.3 22.7 24.2 23.6 25.4 27.2 29.0 30.7	15 x 8	8x8A 7½x9I 5x10N 6x10N
120	26,200	37,050	Single Back To Back	6'0"x8'10" 7'0"x7'10" 7'0"x8'4" 7'0"x8'10" 7'0"x9'4" 7'0"x9'10" 4'0"x6'10" 4'6"x5'10" 4'6"x6'4" 4'6"x6'10" 4'6"x7'4" 5'0"x6'4" 5'0"x6'10" 5'0"x7'4" 5'0"x7'10"	24.2 25.4 27.2 29.0 30.7 32.5 25.2 24.2 26.2 28.4 30.6 28.2 30.8 33.2 35.4	15 x 8	8x8A 7x12N
130	30,550	43,250	Single Back To Back	7'0"x8'10" 7'0"x9'4" 7'0"x9'10" 4'6"x6'10" 4'6"x7'4" 5'0"x6'10" 5'0"x7'4" 5'0"x7'10" 6'0"x7'4" 6'0"x7'10"	29.0 30.7 32.5 28.4 30.6 30.8 33.2 35.4 39.6 42.6	15x8	8x8A 7x12N
140	35,650	50,400	Back To Back	5'0"x7'4" 5'0"x7'10" 6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4"	33.2 35.4 39.6 42.6 45.4 48.4 47.2	16x10	10x8A 7x12N
150	40,900	57,900	Back To Back	6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10" 7'0"x8'4"	39.6 42.6 45.4 48.4 47.2 50.8 54.4	18x12	8x10A 8x12N
160	46,450	65,700	Back To Back	6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10" 7'0"x8'4" 7'0"x8'10" 7'0"x9'4" 7'0"x9'10"	45.4 48.4 47.2 50.8 54.4 58.0 61.4 65.0	18x12	8x10A 9x14N

Niagara Conoidal Fans

With Proper Combinations of Heaters and Engines for Public Buildings and Industrial Installations

Fan Number	Cubic Feet of Air per Minute: 1" Static Pressure	Cubic Feet of Air per Minute: 2" Static Pressure	Buffalo Standard Heater: Arrangement	Buffalo Standard Heater: Size	Buffalo Standard Heater: Clear Area Square Feet	Engine Size: Low Pressure	Engine Size: High Pressure
4	4,895	6,920	Single	3'0"x3'10"	5.2		
4½	6,195	8,750	Single	3'0"x4'4" 3'0"x4'10" 3'0"x5'4"	6.0 6.8 7.6		4x4A 3x3½I
5	7,645	10,820	Single	3'0"x5'4" 3'0"x5'10" 4'0"x5'4"	7.6 8.4 9.7	5x5	5x5A 4x3½I
5½	9,250	13,100	Single	3'0"x5'10" 4'0"x5'4" 4'0"x5'10" 4'0"x6'4" 4'0"x6'10" 4'6"x5'10" 4'6"x6'4"	8.4 9.7 10.7 11.2 12.6 12.1 13.1	6x6	5x5A 4x3½I
6	11,000	15,550	Single	4'0"x5'10" 4'0"x6'4" 4'0"x6'10" 4'6"x5'10" 4'6"x6'4" 4'6"x6'10" 4'6"x7'4"	10.7 11.2 12.6 12.1 13.1 14.2 15.3	8x6	5x5A 4½x5I
7	14,980	21,200	Single	4'6"x6'10" 4'6"x7'4" 5'0"x6'10" 5'0"x7'4" 5'0"x7'10" 6'0"x7'4" 6'0"x7'10"	14.2 15.3 15.4 16.6 17.7 19.8 21.3	10x8	5x5A 5½x7I
8	19,550	27,650	Single	5'0"x7'10" 6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10" 7'0"x8'4"	17.7 19.8 21.3 22.7 24.2 23.6 25.4 27.2	10x8	6x6A 5½x7I
9	24,750	35,050	Single	6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10" 7'0"x8'4" 7'0"x8'10" 7'0"x9'4" 7'0"x9'10"	22.7 24.2 23.6 25.4 27.2 29.0 30.7 32.5	12x8	6x6A 6½x8I
10	30,550	43,250	Single	7'0"x8'4" 7'0"x8'10" 7'0"x9'4" 7'0"x9'10"	27.2 29.0 30.7 32.5	15x8	7x7A 6½x8I
10	30,550	43,250	Back To Back	4'6"x6'10" 4'6"x7'4" 5'0"x6'10" 5'0"x7'4" 5'0"x7'10" 6'0"x7'4" 6'0"x7'10"	28.4 30.6 30.8 33.2 35.4 39.6 42.6	15x8	8x8A 7½x9I 5x10N
11	37,000	52,300	Back To Back	5'0"x7'10" 6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10"	35.4 39.6 42.6 45.4 48.4 47.2 50.8	15x8	8x8A 7½x9I 5x10N
12	44,050	62,300	Back To Back	6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10" 7'0"x8'4" 7'0"x8'10" 7'0"x9'4"	42.6 45.4 48.4 47.2 50.8 54.4 58.0 61.4	15x8	8x8A 10x8A 7x12N
13	51,650	73,050	Back To Back Three Side By Side	7'0"x7'10" 7'0"x8'4" 7'0"x8'10" 7'0"x9'4" 7'0"x9'10" 5'0"x7'10" 6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4"	50.8 54.4 58.0 61.4 65.0 53.1 59.4 63.9 68.1 72.6 70.8	15x10	8x8A 10x8A 7x12N
14	60,000	84,900	Back To Back Three Side By Side	7'0"x8'10" 7'0"x9'4" 7'0"x9'10" 6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10" 7'0"x8'4"	58.0 61.4 65.0 59.4 63.9 68.1 72.6 70.8 76.2 81.6	16x10	8x10A 7x12N

Turbo Conoidal Fans

With Proper Combinations of Heaters and Engines for Public Buildings and Industrial Installations

Fan Number	Cubic Feet of Air per Minute: 1" Static Pressure	Cubic Feet of Air per Minute: 2" Static Pressure	Buffalo Standard Heater: Arrangement	Buffalo Standard Heater: Size	Buffalo Standard Heater: Clear Area Square Feet	Engine: Low Pressure	Engine: High Pressure
4	4,450	6,270	Single	3'0"x3'4" 3'0"x3'10"	4.4 5.2		
4½	5,640	7,950	Single	3'0"x3'10" 3'0"x4'4" 3'0"x4'10" 3'0"x5'4"	5.2 6.0 6.8 7.6		4x4A 4x3½I
5	6,950	9,800	Single	3'0"x4'10" 3'0"x5'4" 3'0"x5'10" 4'0"x5'4"	6.8 7.6 8.4 9.7	5x5	5x5A 4x3½I
5½	8,400	11,880	Single	3'0"x5'10" 4'0"x5'4" 4'0"x5'10" 4'0"x6'4"	8.4 9.7 10.7 11.2	6x6	5x5A 4½x5I
6	10,000	14,120	Single	4'0"x5'4" 4'0"x5'10" 4'0"x6'4" 4'0"x6'10" 4'6"x5'10" 4'6"x6'4"	9.7 10.7 11.2 12.6 12.1 13.1	8x6	5x5A 4½x5I
6½	11,750	16,600	Single	4'0"x6'4" 4'0"x6'10" 4'6"x5'10" 4'6"x6'4" 4'6"x6'10" 4'6"x7'4" 5'0"x6'4" 5'0"x6'10" 5'0"x7'4"	11.2 12.6 12.1 13.1 14.2 15.3 14.1 15.4 16.6	8x6	5x5A 4½x5I
7	13,610	19,250	Single	4'6"x6'4" 4'6"x6'10" 4'6"x7'4" 5'0"x6'4" 5'0"x6'10" 5'0"x7'4" 5'0"x7'10"	13.1 14.2 15.3 14.1 15.4 16.6 17.7	10x8	6x6A 5½x7I
7½	15,610	22,100	Single	5'0"x6'10" 5'0"x7'4" 5'0"x7'10" 6'0"x7'4" 6'0"x7'10"	15.4 16.6 17.7 19.8 21.3	10x8	6x6A 5½x7I
8	17,800	25,100	Single	5'0"x7'4" 5'0"x7'10" 6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4"	16.6 17.7 19.8 21.3 22.7 24.2 23.6	10x8	6x6A 5½x7I
8½	20,100	28,400	Single	6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10" 7'0"x8'4"	19.8 21.3 22.7 24.2 23.6 25.4 27.2	10x8	6x6A 5½x7I 7x7A
			Back To Back	4'0"x5'4" 4'0"x5'10" 4'0"x6'4" 4'0"x6'10" 4'6"x5'10" 4'6"x6'4" 4'6"x6'10"	19.4 21.4 22.4 25.2 24.2 26.2 28.4		
9	22,500	31,800	Single	6'0"x7'10" 6'0"x8'4" 6'0"x8'10" 7'0"x7'4" 7'0"x7'10" 7'0"x8'4" 7'0"x8'10" 7'0"x9'4"	21.3 22.7 24.2 23.6 25.4 27.2 29.0 30.7	12x8	7x7A 6½x8I 6x10N
			Back To Back	4'0"x5'10" 4'0"x6'4" 4'0"x6'10" 4'6"x5'10" 4'6"x6'4" 4'6"x6'10" 4'6"x7'4" 5'0"x6'4" 5'0"x6'10"	21.4 22.4 25.2 24.2 26.2 28.4 30.6 28.2 30.8		
10	27,800	39,300	Single	7'0"x7'10" 7'0"x8'10" 7'0"x9'4"	25.4 29.0 30.7	15x8	7x7A 6½x8I 6x10N
			Back To Back	4'6"x6'4" 4'6"x6'10" 4'6"x7'4" 5'0"x6'4" 5'0"x6'10" 5'0"x7'4" 5'0"x7'10"	26.2 28.4 30.6 28.2 30.8 33.2 35.4		
11	33,700	47,150	Back To Back	5'0"x7'4" 5'0"x7'10" 6'0"x7'4" 6'0"x7'10" 6'0"x8'4" 7'0"x7'4"	33.2 35.4 39.6 42.6 45.4 47.2	15x8	8x8A 7½x9I 6x10N

Buffalo

B. T. U. Transmitted per Hour per Square Foot of Heating Surface For Various Differences in Temperature Between Inside and Outside Air

Material		1°	50	60	70	80	85	90
		Temperature Difference						
Windows, Single Glass, Full Sash Area.	a	1.09	54.5	65.4	76.3	87.2	92.7	98.1
Double Full Glass, Sash Area.	a	0.46	23.0	27.6	32.2	36.8	39.1	41.4
Plate Glass, Full Sash Area	b	1.00	50.0	60.0	70.0	80.0	85.0	90.0
Skylight, Single Glass, Full Sash Area	a	1.16	58.0	69.6	81.2	92.8	98.6	104.4
Double Glass, Full Sash Area.	a	0.48	24.0	28.8	33.6	38.4	40.8	43.2
Doors or **Partitions** Pine 1" Thick	a	0.41	20.5	24.6	28.7	32.8	34.9	36.9
1½" "	a	0.32	16.0	19.2	22.4	25.6	27.2	28.8
2" "	a	0.27	13.5	16.2	18.9	21.6	23.0	24.3
Brick Wall, Plain, 8½" Thick.	a	0.37	18.5	22.2	25.9	29.6	31.5	33.3
13" "	a	0.29	14.5	17.4	20.3	23.2	24.7	26.1
17½" "	a	0.25	12.5	15.0	17.5	20.0	21.3	22.5
22" "	a	0.22	11.0	13.2	15.4	17.6	18.7	19.8
26½" "	a	0.19	9.5	11.4	13.3	15.2	16.1	17.1
¾" Plaster on one side, 8½" Thick.	a	0.36	18.0	21.6	25.2	28.8	30.6	32.6
13" "	a	0.28	14.0	16.8	19.6	22.4	23.8	25.2
17½" "	a	0.24	12.0	14.4	16.8	19.2	20.4	22.6
22" "	a	0.21	10.5	13.6	15.7	17.8	18.9	19.9
26½" "	a	0.18	9.0	10.8	12.6	14.4	15.3	16.2
2.4" Air Space, ¾" Plaster on one side, 8½" Thick.	a	0.25	12.5	15.0	17.5	20.0	21.3	22.5
12½" "	a	0.21	10.5	12.6	15.7	17.8	18.9	19.9
17" "	a	0.19	9.5	11.4	13.3	15.2	16.1	17.1
21½" "	a	0.16	8.0	9.6	11.2	12.8	13.6	14.4
26" "	a	0.14	7.0	8.4	9.8	11.2	11.9	12.6
Furred and ¾" Plaster 4" Thick	a	0.28	14.0	16.8	19.6	22.4	23.8	25.2
8½" "	a	0.23	11.5	13.8	16.1	18.4	19.6	20.7
13" "	a	0.20	10.0	12.0	14.0	16.0	17.0	18.0
17½" "	a	0.18	9.0	10.8	12.6	14.4	15.3	16.2
22" "	a	0.16	8.0	9.6	11.2	12.8	13.6	14.4
Frame Wall, Clapboard, Stud and Plaster. 5" Thick	a	0.44	22.0	26.4	30.8	35.2	37.4	39.6
Clapboard, Paper, Stud and Plaster "	a	0.31	15.5	18.6	21.7	24.8	26.4	27.9
Clapboard, Sheathing, Stud and Plaster.	a	0.28	14.0	16.8	19.6	22.4	23.8	25.2
Clapboard, Paper, Sheathing, Stud and Plaster.	a	0.23	11.5	13.8	16.1	18.4	19.6	20.7
Concrete, Solid, 2" Thick.	b	0.78	39.0	46.8	54.6	62.4	66.3	70.2
3" "	b	0.71	35.5	42.6	49.7	56.8	60.4	63.9
4" "	b	0.66	33.0	39.6	46.2	52.8	56.1	59.4
6" "	b	0.56	28.0	33.6	39.2	44.8	47.6	50.4
Partition, Hollow Tile ½" Plaster both sides 2" Thick	b	0.41	20.5	24.6	28.7	32.8	34.9	36.9
4" "	b	0.33	16.5	19.8	23.1	26.4	28.1	29.7
6" "	b	0.28	14.0	16.8	19.6	22.4	23.8	25.2
Stud, Lath and Plaster on one side.	a	0.60	30.0	36.0	42.0	48.0	51.0	54.0
Lath and Plaster on both sides.	a	0.34	17.0	20.4	23.8	27.2	28.9	30.6
Solid Plaster 2" Thick	b	0.60	30.0	36.0	42.0	48.0	51.0	54.0
3" "	b	0.50	25.0	30.0	35.0	40.0	42.5	45.0
Floor, Single ⅞" no Plaster beneath Joists.	a	0.45	22.5	27.0	31.5	36.0	38.3	40.5
Lath and Plaster beneath Joists.	a	0.26	13.0	15.6	18.2	20.8	22.1	23.4
Double 1⅞" no Plaster beneath Joists.	a	0.31	15.5	18.6	21.7	24.8	26.4	27.9
Lath and Plaster beneath Joists.	a	0.18	9.0	10.8	12.6	14.4	15.3	16.2
Single on Brick Arch.	b	0.15	7.5	9.0	10.5	12.0	12.8	13.5
Fireproof construction.	b	0.10	5.0	6.0	7.0	8.0	8.5	9.0
Concrete on Brick Arch.	b	0.20	10.0	12.0	14.0	16.0	17.0	18.0
Laid on Ground								
Cement or Tile laid on ground no Wood above.	a	0.31	15.5	18.6	21.7	24.8	26.4	27.9
" " " " " " Wood Floor above	a	0.10	5.0	6.0	7.0	8.0	8.5	9.0
Dirt, no Floor whatever	a	0.20	10.0	12.0	14.0	16.0	17.0	18.0
Roof. Tile 1" Thick	a	0.80	40.0	48.0	56.0	64.0	68.0	72.0
Hollow Tile, 6" thick, Concrete 8" thick, tar and gravel	a	0.35	17.5	21.0	24.5	28.0	29.8	31.5
Slate, on 1" Planks.	a	0.43	21.5	25.8	30.1	34.4	36.6	38.7
" on Wooden Framing.	a	0.85	42.5	51.0	59.5	68.0	72.3	76.5
Tar, Paper and Gravel on 2" Planks.	a	0.26	13.0	15.6	18.2	20.8	22.1	23.4
Sheet Iron.	a	1.20	60.0	72.0	84.0	96.0	102.0	108.0
Corrugated Iron.	a	1.50	75.0	90.0	105.0	120.0	127.5	135.0
Concrete with Cinder Fill. 2" Thick.	a	0.80	40.0	48.0	56.0	64.0	68.0	72.0
4" "	a	0.60	30.0	36.0	42.0	48.0	51.0	54.0
6" "	a	0.54	27.0	32.4	37.8	43.2	45.9	48.6
Asbestos Shingles on 1" Tongue and Groove Boards.	c	0.30	15.0	18.0	21.0	24.0	25.5	27.0
Ajax Built up Roofing, 3 ply, on 4" Concrete with two layers	c	0.508	25.4	30.5	35.6	40.6	43.2	45.7
" " " " of Double Neptune Felt.	c	0.303	15.2	18.2	21.2	24.2	25.8	27.3
Phoenix Built up Roofing, 4 ply, on 2" Boards, with two layers.	c	0.30	15.0	18.0	21.0	24.0	25.5	27.0
" " " " of Double Neptune Felt.	c	0.21	10.5	12.6	14.7	16.8	17.9	18.9

Authority:—a, Buffalo Forge Company. b, American Society of Heating and Ventilating Engineers. c, Johns-Manville Company.

Buffalo

Properties of Dry Air

Barometric Pressure 29.921 Inches

Temperature Degrees Fahr.	Weight per Cu. Ft. Pounds	Per Cent. of Volume at 70° F.	B. T. U. Absorbed by one Cu. Ft. Dry Air per Degree F.	Cu. Ft. Dry Air Warmed One degree per B. T. U.	Temperature Degrees Fahr.	Weight per Cu. Ft. Pounds	Per Cent. of Volume at 70° F.	B. T. U. Absorbed by One Cu. Ft. Dry Air per Degree F.	Cu. Ft. Dry Air Warmed One Degree per B. T. U.
0	.08636	.8680	.02080	48.08	130	.06732	1.1133	.01631	61.32
5	.08544	.8772	.02060	48.55	135	.06675	1.1230	.01618	61.81
10	.08453	.8867	.02039	49.05	140	.06620	1.1320	.01605	62.31
15	.08363	.8962	.02018	49.56	145	.06565	1.1417	.01592	62.82
20	.08276	.9057	.01998	50.05	150	.06510	1.1512	.01578	63.37
25	.08190	.9152	.01977	50.58	160	.06406	1.1700	.01554	64.35
30	.08107	.9246	.01957	51.10	170	.06301	1.1890	.01530	65.36
35	.08025	.9340	.01938	51.60	180	.06205	1.2080	.01506	66.40
40	.07945	.9434	.01919	52.11	190	.06110	1.2270	.01484	67.40
45	.07866	.9530	.01900	52.64	200	.06018	1.2455	.01462	68.41
50	.07788	.9624	.01881	53.17	220	.05840	1.2833	.01419	70.48
55	.07713	.9718	.01863	53.68	240	.05673	1.3212	.01380	72.46
60	.07640	.9811	.01846	54.18	260	.05516	1.3590	.01343	74.46
65	.07567	.9905	.01829	54.68	280	.05367	1.3967	.01308	76.46
70	.07495	1.0000	.01812	55.19	300	.05225	1.4345	.01274	78.50
75	.07424	1.0095	.01795	55.72	350	.04903	1.5288	.01197	83.55
80	.07356	1.0190	.01779	56.21	400	.04618	1.6230	.01130	88.50
85	.07289	1.0283	.01763	56.72	450	.04364	1.7177	.01070	93.46
90	.07222	1.0380	.01747	57.25	500	.04138	1.8113	.01018	98.24
95	.07157	1.0472	.01732	57.74	550	.03932	1.9060	.00967	103.42
100	.07093	1.0570	.01716	58.28	600	.03746	2.0019	.00923	108.35
105	.07030	1.0660	.01702	58.76	700	.03423	2.1900	.00847	118.07
110	.06968	1.0756	.01687	59.28	800	.03151	2.3785	.00782	127.88
115	.06908	1.0850	.01673	59.78	900	.02920	2.5670	.00728	137.37
120	.06848	1.0945	.01659	60.28	1000	.02720	2.7560	.00680	147.07
125	.06790	1.1040	.01645	60.79	1200	.02392	3.1335	.00603	165.83

Properties of Saturated Steam

Temperature F.	Approximate Gauge Pressure	Density	Specific Volume Cubic Feet Per Pound	Heat of Liquid B. T. U.	Latent Heat B. T. U.	Total Heat B. T. U.
212	0	0.03732	26.79	180.0	970.4	1150.4
215	1	0.03945	25.35	183.0	968.4	1151.5
219	2	0.04243	23.57	187.1	965.9	1152.9
222	3	0.04477	22.34	190.1	963.9	1154.0
224	4	0.04640	21.55	192.1	962.6	1154.8
227	5	0.04892	20.44	195.2	960.7	1155.8
230	6	0.0516	19.39	198.2	958.7	1156.9
232	7	0.0534	18.72	200.2	957.4	1157.6
235	8	0.0562	17.78	203.2	955.4	1158.7
237	9	0.0582	17.17	205.3	954.1	1159.4
239	10	0.0602	16.60	207.3	952.8	1160.0
250	15	0.0724	13.82	218.5	945.3	1163.8
259	20	0.0837	11.95	227.6	939.1	1166.7
267	25	0.0949	10.54	235.8	933.5	1169.3
274	30	0.1057	9.46	242.9	928.6	1171.5
281	35	0.1174	8.51	250.1	923.5	1173.6
287	40	0.1283	7.79	256.2	919.1	1175.3
298	50	0.1504	6.65	267.5	911.0	1178.5
307	60	0.1707	5.86	276.8	904.2	1181.0
316	70	0.1930	5.19	286.1	897.3	1183.3
324	80	0.2148	4.66	294.3	891.0	1185.4
331	90	0.2353	4.250	301.6	885.5	1187.1
338	100	0.2575	3.884	308.9	879.9	1188.8
344	110	0.2778	3.600	315.1	875.1	1190.2
350	120	0.2992	3.342	321.4	870.1	1191.5
356	134	0.3221	3.105	327.7	865.2	1192.9
361	140	0.3423	2.922	332.9	861.0	1193.9

Condensed from Marks and Davis Steam Tables.

FAN SYSTEM OF HEATING, VENTILATING AND HUMIDIFYING

Weight per Lineal Foot for Galvanized Iron Pipes

U. S. Standard Gauge

Diameter of Pipe	Square Feet per Running Foot	Number of Gauge 26	24	22	20	18	16
4	1.13	1.13	1.47	1.69	1.97	2.56	3.10
5	1.39	1.39	1.80	2.08	2.43	3.19	3.82
6	1.65	1.65	2.14	2.47	2.89	3.79	4.53
7	1.91	1.91	2.48	2.86	3.34	4.39	5.27
8	2.18	2.18	2.83	3.27	3.81	5.00	6.00
9	2.44	2.44	3.17	3.66	4.27	5.61	6.71
10	2.70	2.70	3.51	4.05	4.72	6.21	7.42
11	2.96	2.96	3.85	4.44	5.18	6.80	8.14
12	3.22	3.22	4.18	4.83	5.63	7.40	8.87
13	3.48	3.48	4.52	5.22	6.09	8.00	9.57
14	3.74	3.74	4.86	5.61	6.54	8.60	10.29
15	4.01	4.01	5.21	6.01	7.01	9.22	10.96
16	4.27	4.27	5.55	6.40	7.47	9.82	11.73
17	4.53	4.53	5.89	6.79	7.92	10.42	12.45
18	4.87	4.87	6.33	7.30	8.51	11.18	[illegible]
19	5.14	5.14	6.68	7.71	9.00	11.80	14.11
20	5.40	5.40	7.02	8.10	9.45	12.42	14.87
21	5.66	5.66	7.26	8.49	9.78	12.85	[illegible]
22	5.92	5.92	7.70	8.88	10.35	13.60	16.25
23	6.18	6.18	8.04	9.27	10.81	14.10	17.00
24	6.45	6.45	8.38	9.67	11.30	14.84	17.71
25	6.71	6.71	8.72	10.06	11.74	15.41	18.41
26	6.97	6.97	9.05	10.45	12.20	16.00	19.12
27	7.33	7.33	9.40	10.85	12.67	16.62	19.87
28	7.59	7.59	9.75	11.27	13.13	17.26	20.60
29	7.75	7.75	10.07	11.63	13.58	17.81	[illegible]
30	8.10	8.10	10.54	12.17	14.20	18.62	22.25
31	8.36	8.36	10.87	12.54	14.63	19.20	23.00
32	8.62	8.62	11.20	12.93	15.10	19.84	23.70
33	8.88	8.88	11.56	13.34	15.56	20.42	24.40
34	9.15	9.15	11.90	13.73	16.00	21.08	25.18
35	9.41	9.41	12.23	14.10	16.48	21.65	25.85
36	9.67	9.67	12.57	14.50	16.91	22.22	26.65
37	9.93	9.93	12.91	14.90	17.40	22.84	27.50
38	10.19	10.19	13.25	15.29	17.81	23.40	28.40
39	10.46	10.46	13.60	15.60	18.31	24.02	28.70
40	10.72	10.72	13.95	16.08	18.76	24.68	29.70
41	10.98	10.98	14.27	16.47	19.20	25.25	30.2
42	11.24	11.24	14.60	16.86	19.61	25.86	30.90
43	11.50	11.50	15.06	17.38	20.30	26.60	31.80
44	11.85	11.85	15.40	17.78	20.74	27.25	32.60
45	12.11	12.11	15.75	18.17	21.29	27.90	33.30
46	12.37	12.37	16.10	18.55	21.62	28.43	34.00
47	12.63	12.63	16.40	18.95	22.10	29.00	34.70
48	12.90	12.90	16.78	19.35	22.60	29.70	35.50
49	13.15	13.15	17.10	19.72	23.00	30.25	36.20
50	13.41	13.41	17.45	20.12	23.50	30.90	36.90
51	13.66	13.66	17.76	20.49	23.90	31.40	37.50
52	13.94	13.94	18.12	20.97	24.40	32.00	[illegible]
53	14.20	14.20	18.46	21.30	24.90	32.66	39.00
54	14.46	14.46	18.80	21.69	25.30	33.20	39.70
55	14.81	14.81	19.28	22.22	25.94	34.10	40.8
56	15.07	15.07	19.60	22.61	26.40	34.65	41.40
57	15.33	15.33	19.95	23.00	26.80	35.24	42.10
58	15.58	15.58	20.30	23.37	27.30	35.84	42.8
59	15.83	15.83	20.55	23.74	27.70	36.40	43.5
60	16.12	16.12	20.95	24.18	28.21	37.00	45.3
62	16.65	16.65	21.65	24.97	29.10	38.20	45.70
64	17.16	17.16	22.30	25.74	30.00	39.50	47.2
66	17.66	17.66	22.97	26.49	30.90	40.60	48.50
68	18.21	18.21	23.65	27.31	31.83	41.80	[illegible]
70	18.73	18.73	24.40	28.12	32.80	43.10	51.50
72	19.25	19.25	25.02	28.92	33.70	44.30	[illegible]
74	19.79	19.79	25.70	29.68	34.65	45.50	[illegible]
76					35.62	45.77	[illegible]
78					35.75	46.96	55.1
80					36.65	48.16	[illegible]
82					37.57	49.40	[illegible]
84					38.50	50.60	[illegible]
86					39.39	51.77	60.77
Weight per Square Foot		1.00	1.30	1.50	1.75	2.30	2.70

Heating and Ventilating
Ducts to 19" Dia. 26 Ga.
" 20" to 29" " 24 "
" 30" to 39" " 22 "
" 40" to 49" " 20 "
" 50" to 84" " 18 "
Above 84" " 16 "

For Planing-Mill Work
Ducts to 8" Dia. 24 Ga.
" 9" to 14" " 22 "
" 15" to 20" " 20 "
" 21" to 30" " 18 "

Weights in Lbs. (Avoir.) per Running Ft.

Weight of Black Steel Pipes in Pounds (Avor.) per Running Foot

Dia. Pipe	Material Sq. Ft. per Running Ft.	NUMBER OF GAUGE, U. S. S.						
		No. 24	No. 22	No. 20	No. 18	No. 16	No. 14	No. 12
4	1.13	1.30	1.58	1.86	2.43	2.99	3.62	5.08
5	1.39	1.60	1.95	2.29	2.99	3.68	4.45	6.25
6	1.65	1.90	2.31	2.72	3.54	4.36	5.28	7.42
7	1.91	2.20	2.67	3.15	4.10	5.05	6.11	8.58
8	2.18	2.50	3.05	3.60	4.68	5.77	6.97	9.80
9	2.44	2.80	3.42	4.03	5.25	6.47	7.80	10.98
10	2.70	3.10	3.78	4.45	5.80	7.15	8.64	12.15
11	2.96	3.40	4.15	4.88	6.36	7.85	9.47	13.31
12	3.22	3.70	4.50	5.31	6.91	8.52	10.30	14.48
13	3.48	4.00	4.88	5.74	7.48	9.21	11.15	15.66
14	3.74	4.30	5.23	6.17	8.03	9.90	11.97	16.84
15	4.01	4.61	5.61	6.61	8.61	10.61	12.83	18.03
16	4.27	4.91	5.97	7.04	9.16	11.29	13.65	19.17
17	4.53	5.21	6.35	7.48	9.74	12.00	14.49	20.40
18	4.87	5.60	6.81	8.03	10.45	12.89	15.55	21.90
19	5.14	5.91	7.20	8.48	11.04	13.60	16.42	23.10
20	5.40	6.21	7.56	8.90	11.60	14.30	17.26	24.30
21	5.59	6.43	7.83	9.22	12.00	14.80	17.87	25.10
22	5.92	6.80	8.28	9.75	12.70	15.65	18.90	26.60
23	6.18	7.11	8.66	10.20	13.29	16.38	19.80	27.80
24	6.45	7.41	9.04	10.63	13.85	17.08	20.65	29.00
25	6.71	7.71	9.40	11.06	14.40	17.75	21.50	30.20
26	6.97	8.01	9.75	11.48	14.96	18.41	22.30	31.30
27	7.23	8.31	10.11	11.93	15.51	19.12	23.10	32.50
28	7.50	8.62	10.50	12.38	16.10	19.87	24.00	33.75
29	7.75	8.91	10.85	12.78	16.67	20.50	24.80	34.90
30	8.10	9.32	11.34	13.37	17.40	21.45	25.90	36.40
31	8.36	9.61	11.70	13.80	18.00	22.15	26.75	37.60
32	8.62	9.92	12.07	14.25	18.52	22.83	27.60	38.80
33	8.88	10.21	12.45	14.66	19.10	23.50	28.40	40.00
34	9.15	10.53	12.81	15.10	19.68	24.43	29.30	41.20
35	9.41	10.82	13.18	15.51	20.20	24.90	30.10	42.30
36	9.67	11.11	13.54	15.95	20.78	25.60	30.90	43.50
37	9.93	11.42	13.90	16.40	21.38	26.30	31.80	44.70
38	10.19	11.71	14.28	16.80	21.90	27.00	32.60	45.80
39	10.46	12.03	14.65	17.27	22.50	27.74	33.50	47.10
40	10.72	12.33	15.00	17.70	23.01	28.40	34.30	48.25
41	10.98	12.62	15.38	18.11	23.60	29.10	35.10	49.40
42	11.24	12.93	15.75	18.55	24.20	29.80	36.00	50.60
43	11.59	13.32	16.21	19.10	24.90	30.70	37.05	52.10
44	11.85	13.64	16.60	19.55	25.50	31.40	37.90	53.30
45	12.11	13.93	16.97	20.00	26.00	32.10	38.75	54.50
46	12.37	14.23	17.31	20.40	26.60	32.80	39.60	55.70
47	12.63	14.52	17.70	20.85	27.20	33.45	40.40	56.80
48	12.90	14.83	18.07	21.30	27.75	34.20	41.30	58.00
49	13.15	15.11	18.40	21.70	28.25	34.80	42.10	59.20
50	13.41	15.42	18.80	22.15	28.80	35.55	42.90	60.40
51	13.66	15.71	19.13	22.55	29.40	36.20	43.75	61.50
52	13.94	16.01	19.50	23.00	30.00	36.90	44.60	62.65
54	14.46	16.62	20.25	23.85	31.10	38.30	46.30	65.00
56	15.07	17.32	21.10	24.85	32.40	39.90	48.20	67.80
58	15.58	17.91	21.80	25.70	33.50	41.30	49.80	70.20
60	16.12	18.53	22.60	26.65	34.70	42.75	51.60	72.60
62	16.65	19.16	23.30	27.50	35.80	44.10	53.30	75.00
64	17.16	19.72	24.00	28.30	36.90	45.50	54.90	77.20
66	17.66	20.30	24.70	29.15	38.00	46.80	56.50	79.40
68	18.21	20.95	25.50	30.00	39.15	48.25	58.30	81.80
70	18.75	21.55	26.25	30.90	40.30	49.70	60.00	84.30
72	19.25	22.15	27.00	31.80	41.40	51.00	61.60	86.60
74	19.79	22.75	27.70	32.65	42.60	52.40	63.30	89.00

Carrying Capacity of Pipes

This table specifies the diameters of pipes required for the passage of stated volumes of air at given velocities. The column, "Cubic feet of air per minute," indicates various quantities of air to be moved per minute. The figures at top of table give the velocities in feet per minute at which the air is to be moved, and the figures in the body of the table state the required diameters of pipes for the passage of the volumes mentioned at the given velocities.

DIAMETER OF PIPE IN INCHES

Cubic Feet of Air per Minute	VELOCITIES												Cubic Feet of Air per Minute	VELOCITIES								
	500	600	800	1000	1200	1500	1800	2000	2500	3000	3500	4000		1000	1200	1500	1800	2000	2500	3000	3500	4000
200	9	8	7	7	6	6	6	6	6	6	6	6	31000	76	69	62	57	54	48	44	41	38
400	13	11	10	9	8	8	7	7	6	6	6	6	32000	77	70	63	57	55	49	45	41	39
600	15	14	12	11	10	9	8	8	7	7	6	6	33000	78	72	64	58	56	50	45	42	39
800	18	16	14	13	12	10	9	9	8	8	7	7	34000	79	73	65	59	56	50	46	43	40
1000	20	18	16	14	13	12	10	10	9	8	8	7	35000	81	74	66	60	57	51	47	43	40
1200	21	20	17	15	14	13	11	11	10	9	9	8	36000	82	75	67	61	58	52	47	44	41
1400	23	21	18	16	15	14	12	12	11	10	9	9	37000	83	76	68	62	59	52	48	44	42
1600	25	23	20	18	16	15	13	13	11	11	10	9	38000	84	77	69	63	60	53	49	45	42
1800	26	24	21	19	17	15	14	13	12	11	10	10	39000	85	78	70	63	60	54	49	46	43
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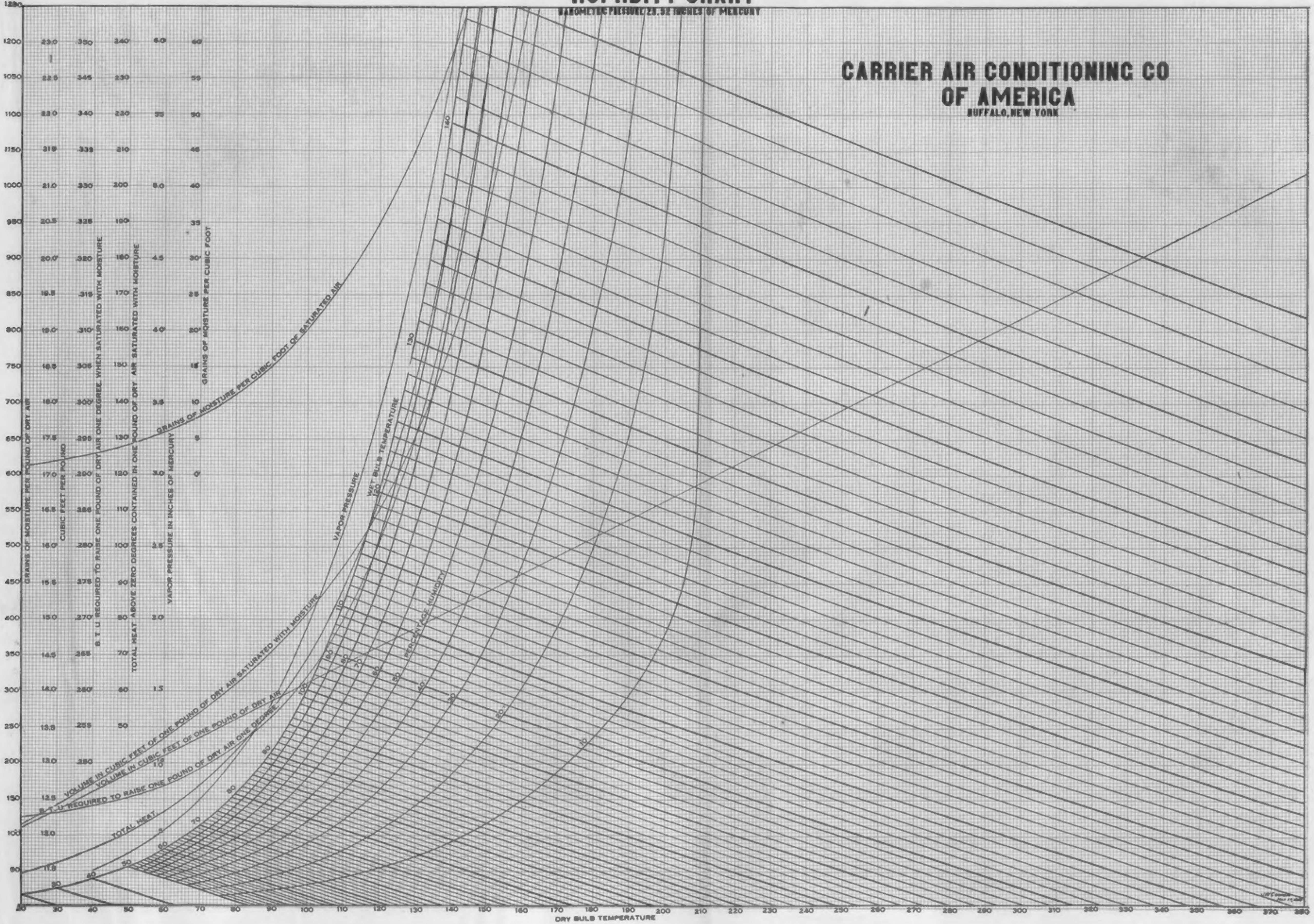
30000				75	68	61	56	53	47	43	40	38	100000					92	86	79	73	68

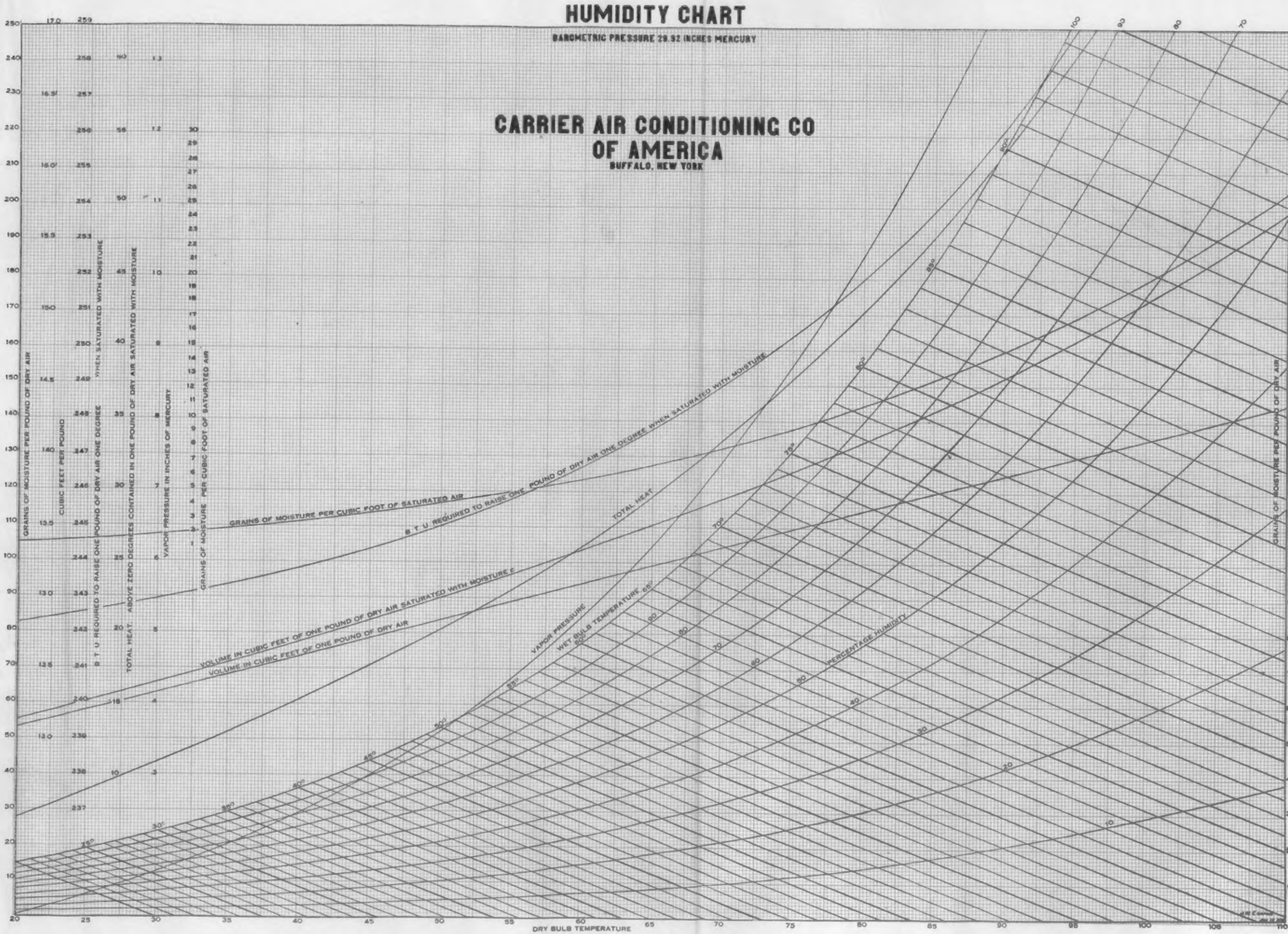
INDEX

INDEX—Continued

HUMIDITY CHART

BAROMETRIC PRESSURE, 29.92 INCHES OF MERCURY




HUMIDITY CHART
BAROMETRIC PRESSURE 29.92 INCHES MERCURY
CARRIER AIR CONDITIONING CO
OF AMERICA
BUFFALO, NEW YORK
GRAINS OF MOISTURE PER POUND OF DRY AIR
CUBIC FEET PER POUND
B. T. U. REQUIRED TO RAISE ONE POUND OF DRY AIR ONE DEGREE WHEN SATURATED WITH MOISTURE
TOTAL HEAT, ABOVE ZERO DEGREES CONTAINED IN ONE POUND OF DRY AIR SATURATED WITH MOISTURE
VAPOR PRESSURE IN INCHES OF MERCURY
GRAINS OF MOISTURE PER CUBIC FOOT OF SATURATED AIR
GRAINS OF MOISTURE PER CUBIC FOOT OF SATURATED AIR
B. T. U. REQUIRED TO RAISE ONE POUND OF DRY AIR ONE DEGREE WHEN SATURATED WITH MOISTURE
TOTAL HEAT
VOLUME IN CUBIC FEET OF ONE POUND OF DRY AIR SATURATED WITH MOISTURE
VOLUME IN CUBIC FEET OF ONE POUND OF DRY AIR
VAPOR PRESSURE
WET BULB TEMPERATURE
PERCENTAGE HUMIDITY
DRY BULB TEMPERATURE